INTERNAL COOLING OF A HIGH SPEED SUPERCHARGED DIESEL ENGINE BY MANIFOLD WATER INJECTION

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ABSTRACT

INTERNAL COOLING OF A HIGH SPEED SUPERCHARGED DIESEL ENGINE BY MANIFOLD WATER INJECTION

by

Frederick W. Hamilton

Submitted to the Department of Ocean Engineering on 14 May, 1971, in partial fulfillment of the requirements for the degrees of Master of Science in Naval Architecture and Marine Engineering and Master of Science in Mechanical Engineering.

The effect of manifold water injection on the performance characteristics of a high speed diesel was investigated using the 4" bore geometrically similar diesel engine in the Sloan Laboratory. The major emphasis was on the degree of reduction in heat rejection by the engine with water injection.

The engine was run at three speeds and two inlet air pressures over a range of fuel-air ratios with five different water-fuel ratios. Power and heat transfer data were recorded at each operating point at optimum fuel injection advance.

It was found that the heat rejected by the cylinder head decreased with increasing water-fuel ratios. Two models are proposed to explain the phenomenon of internal cooling and two prediction methods are presented to determine the amount of heat rejection decrease with water injection.

Further investigations on internal cooling are recommended primarily to determine the extent of possible reductions in emissions of oxides of nitrogen.

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Thesis Reader: A. D. Carmichael
Professor of Power Engineering
Department of Ocean Engineering

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ACKNOWLEDGEMENTS

I am indebted to all those who helped to make this project possible.

First and foremost I thank my advisor, Professor A. R. Rogowski, whose guidelines and ideas, together with his understanding and interest, made this a most valuable experience for me.

I was very fortunate to have excellent assistance from the staff at the Sloan Automotive Laboratory. Both Larry Johnson, mechanic, and Sam Marcolongo, machinist, aided the project with skill and goodwill. The other students, Dick Vanderpoel and Frank Pompei also contributed by their interest, advice, and encouragement.

Joe Rife, friend and consultant, helped without measure. His expert advice and personal encouragement inspired me in many ways and is most appreciated.

Finally I thank my wife, Bonnie, for her patience and understanding, not only during the course of this investigation, but also during the last two years while I was studying here at M.I.T.

F.M. Hamilton

May 1971

CHAPTER I

INTRODUCTION

The advantages of supercharging a diesel engine have been well established for many years. Briefly,supercharging increases the usable power output, provides better specific fuel consumption, improves torque at low speed and removes smoke from the exhaust. With these advantages, however, there are also disadvantages. Designers of modern over the road truck diesels usually find the limit to increased power output by supercharging to be the amount of heat an acceptably sized cooling system can dissipate. That is to say, overheating of the diesel engine in this application is the limit first reached in possible power output.

Since intake manifold water injection has been used successfully for many years in spark ignition engines as an internal coolant to supress combustion knock and more recently to reduce flame temperature in an effort to reduce nitric oxide emissions, it appears reasonable to conclude that water could also be used as an internal coolant for diesel engines. In fact, the same properties of water, namely a relatively low vapor pressure at normal induction temperatures, a high latent heat of vaporization and a low boiling point relative to the maximum engine cycle temperatures, used to advantage in the spark ignition engine are the same properties desired of an

internal coolant for a compression ignition engine.

Due to the differences in combustion processes between these two types of engines however, the well published performance data of spark ignition engines operating with intake manifold water injection is of little use in the prediction of diesel performance with internal cooling. Thus, this investigation was conducted to determine the effect of manifold water injection on the performance characteristics of a high speed diesel engine with particular emphasis on the degree of reduction, if any, of the heat rejected by the engine.

The possibility also exists that, as in spark ignition engines, manifold water injection applied to diesel engines may decrease exhaust emissions of oxides of nitrogen and/or further reduce the exhaust smoke. This investigation, however, did not include exhaust gas analysis and this area of performance was not considered.

CHAPTER II

APPARATUS & PROCEDURE

A. THE ENGINE

The engine used in this study was the 4" bore, $2\frac{1}{2}$"stroke geometrically similar diesel engine in the Sloan Laboratory. The basic set-up is the same as was used by DUNCAN and LOGTERMAN in their experiments on heat transfer effects, and utilizes the hemispherical piston they designed in the course of their investigation. The only changes in the set-up were the addition of a water spray system and a super-charging system, and the deletion of the indicator. A list of engine specifications appears in Table II-1.

B. FUEL INJECTION SYSTEM

Roosa Master fuel injection equipment was used on the engine. The injection pump has two opposed plungers which rotate at half engine speed inside a fixed cam. A metered quantity of fuel is pumped into a cavity between the plungers and when the plungers ride over the cam lobes, this fuel is forced through lines to the nozzles.

The nozzles are a very compact type having a low recip-rocating mass and thus having little tendency to produce secondary injections. The nozzle opening pressure was set at 2500 psi.

The injection timing was controlled by means of a plane-tary phase changer mounted between the engine and the fuel

pump. This phase changer both reverses the direction of shaft rotation and reduces the shaft speed by a factor of two, hence the fuel pump was driven directly from the engine crankshaft.

Actual injection advance was calibrated against the markings on the phase changer using automotive breaker points and a strobelight while the injector sprayed fuel into a glass jar. In this manner the start of injection could be seen through the glass and the angle read on the engine's flywheel.

No. 2 Marine Diesel having a cetane number of about 44 was used in this investigation. Table II-2 lists the fuel injection specifications.

C. WATER INJECTION SYSTEM

Before finding a suitable spray nozzle to inject water into the intake manifold several attempts were made at making nozzles from Polyflow tubing fittings by silver soldering one end of the fitting closed and drilling a small orifice in the closed end. These nozzles were unsatisfactory at the low flow rates as a continuous controllable spray could not be achieved. Flashing the water into steam by means of a heat exchanger just prior to the nozzle did not solve the problems.

Finally, the author decided to use a pneumatic spray nozzle manufactured by Spray Engineering Company of Burlington,

TABLE II - 1

ENGINE SPECIFICATIONS

No.		
1	bore	4 in
2.	stroke	$2\frac{1}{2}$ in
3.	cylinder displacement	31.41 in^3
4.	conn rod length	6.25 in
5.	combustion surface	7.65 in^2
6.	squish height	.040 in
7.	clearance volume	2.02 in^3
8.	combustion volume	1.61 in^3
9.	number of compression and oil rings	2,1
10.	number of inlet and exhaust valves	2,2
11.	valve diameter	1.286 in
12.	valve lift	.280 in
13.	inlet valve timing opens-closes	15° BTC - 50° ABC
14.	exhaust valve timing opens-closes	50° BBC - 15° ATC
15.	diameter of intake manifold pipe	2.00 in
16.	diameter of exhaust manifold pipe	1.60 in
17.	abs. oil viscosity	8.69 #/sec-ft
18.	stroke/bore ratio	.625
19.	type of engine cycle	4 stroke

Mass. This was the only method found to achieve satisfactory atomization of the water spray and control of the flow at the low values desired.

The nozzle used in the investigation was SPRAYCO model No. 125M, Set F2070. This nozzle incorporates a control valve to regulate the flow from zero up to several gallons per hour. Designed to operate on a slight suction lift for low flows, or under a pressure head for larger flows, experimentation showed that the best control and spray was achieved by using city water at main pressure (60 psi) for the supply. This was attributed to the extraordinary length of ¼" Polyflow tubing between the flow measuring instrumentation and the spray nozzle.

The nozzle was mounted on the intake manifold oriented so as to spray into the manifold at right angles to the flow of air several inches upstream of the intake valves. Figure II-1 illustrates the spray nozzle.

D. COMPRESSED AIR SYSTEM

The supercharging air compressor in the pump room of the Sloan Laboratory was used to supply compressed air for the pneumatic spray nozzle and the supercharging air to the engine. This air compressor cycles between approximately 90 psi and 120 psi, however the supply to the lab mains is regulated from the storage tank by a high volume flow, low pressure drop regulator designed by MELTON in 1966. This

TABLE II - 2

FUEL INJECTION SPECIFICATIONS

No.	Item	
1.	max fuel per cycle	40 mm^3
2.	no. of nozzle sprays	8
3.	diameter of nozzle holes	.008 in
4.	length of nozzle holes	.020 in
5.	angle of spray from horz.	7°
6.	length of nozzle, inlet stud to tip	3.65 in
7.	diameter of nozzle body	.375 in
8.	I.D. of inlet stud	.100 in
9.	nozzle cracking pressure	2500 psi
10.	length of injection lines	41 in
11.	I.D. & O.D. injection line	.062, .250 in
12.	fuel pump model number	DBGF C229-X2B
13.	inj. pump plunger dia	.290 in
14.	max plunger lift	.0186 in
15.	max cam duration crank degrees	19.5°

control regulates the air to within 2 psi at a nominal 100 psi.

The air was further reduced in pressure in the engine cell to approximately 50 psi before final regulation. The final controllable regulation provided air at a nominal 5 psi for the supercharging and at 25 psi to 35 psi for the spray nozzle. The resulting supercharging air fluctuated only about .8" Hg at the engine.

E. INSTRUMENTATION

(1) Fuel Flow

Fuel consumption was measured gravimetrically using the system shown in Figure II-2. The system's only outlet is the fuel injected into the engine; all other lines return to the glass beaker. Additional fuel is added to the beaker between fuel measuring cycles by drawing from the fuel tank. The time required to use a convenient weight of fuel is measured by an electrically tripped timer.

(2) Air Flow

The engine's air consumption was measured by an ASME square-edged orifice with flange taps. Manometers were connected to these taps to indicate static pressure and the pressure drop across the orifice. The orifice diameter used was .614".

An air rotometer was installed in the air supply line to the spray nozzle. The air supplied to the intake manifold by the spray nozzle was typically less than 2% of the engine's air consumption.

(3) Engine Power and Speed

The dynamometer used on the engine is a direct current machine driven at half engine speed. Brake load is transmitted to a hydraulic cylinder which reads out as a column of mercury in a manometer. The single constant of 3.15 converts inches of mercury into BMEP in psi.

Engine speed is measured by a full speed tachometer drive from a right angle drive mounted on the shaft connecting the fuel pump to the crankshaft. This tachometer is used while changing speeds. To insure steady speed during a run, a stroboscope firing on line frequency is aimed at the flywheel where a series of markings come into view every 25 RPM

(4) Temperatures and Water Flow Rates

Iron-constantan thermocouples are installed for measuring oil temperature, head, barrel and shunt, inlet and outlet temperatures, and air inlet before and after water injection. These thermocouples read out on a dial potentiometer manufactured by Brown.

Shunt and barrel cooling water flows and manifold water injection flow were measured by rotometers. These were calibrated by the author prior to the start of the experiment and the calibration curves appear in the Appendix.

(5) Pressures

Mercury manometers were used to measure inlet, exhaust and static upstream air orifice pressure. Barometric

WATER INJECTION SPRAY VALVE

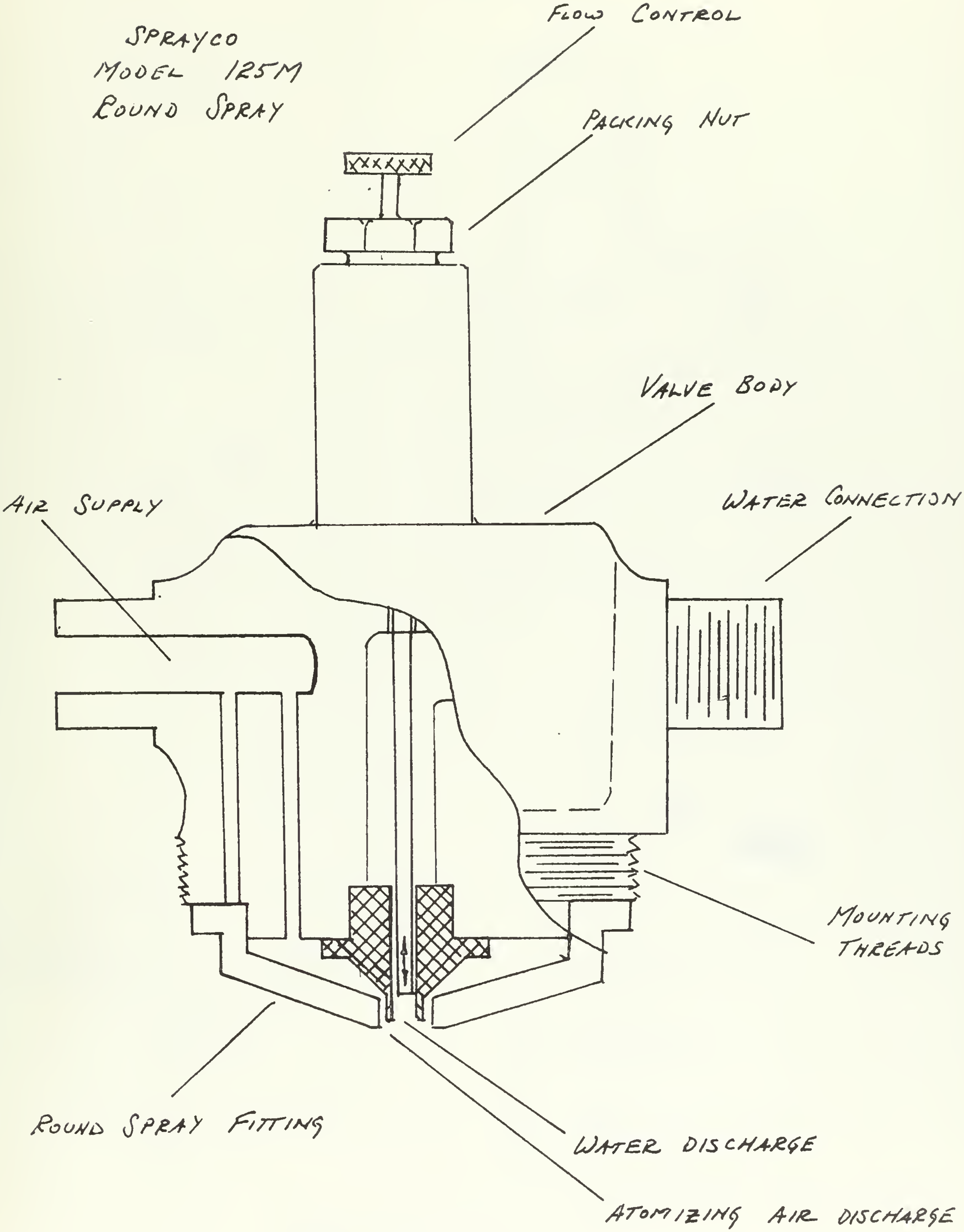


FIGURE II - 1

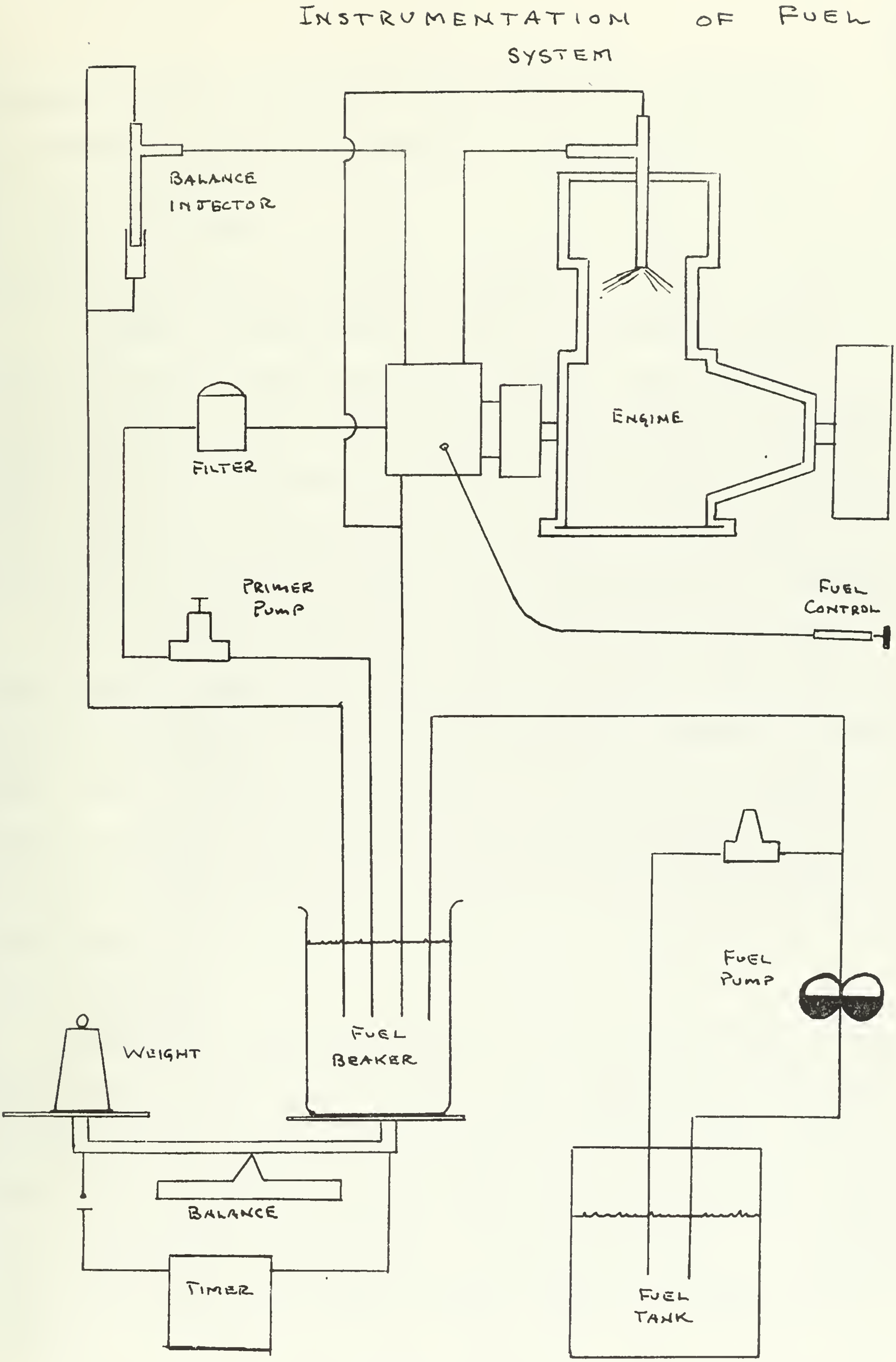

INSTRUMENTATION OF FUEL
SYSTEM
BALANCE
INJECTOR
FILTER
ENGINE
PRIMER
PUMP
FUEL
CONTROL
FUEL
PUMP
WEIGHT
FUEL
BEAKER
BALANCE
TIMER
FUEL
TANK

FIGURE II - 2

pressure was not recorded. Oil pressure was registered on a Bourdon gauge on the engine's control panel.

G. TEST PROCEDURE

(1) Data Sought and Variables Examined

In order to determine the effect of manifold water injection on the operation of the engine, the primary characteristics determined were the heat rejected, the power output and the indicated efficiency. In addition, volumetric efficiency was determined as well as data on the change in optimum fuel injection advance with increased water injected.

The operating conditions which were varied included engine speed, fuel flow rate, water-fuel (W/F) ratio and injection advance. Two inlet pressures were examined, atmospheric and 10" Hg above atmospheric. This was to compare effects between unsupercharged and supercharged operation. The lack of time prevented operation at more supercharged pressures.

(2) Conditions

Water-fuel (W/F) ratios of 0, .25, .50, .75 and 1.0 were run at each of three fuel flows, lean (fuel-air, F/A, ratio of about .025), medium (F/A of about .040) and rich (F/A of about .053), at three speeds, 1800, 2400, and 3000 RPM, for the unsupercharged operation. At each W/F ratio the optimum injection advance was found and the operating conditions recorded.

For the supercharged operation, the same W/F ratios were run at the same fuel flow rates and speeds as in the unsupercharged condition. In addition, approximately the same F/A ratios of lean, medium and rich were run. As in the unsupercharged tests, the operating conditions were recorded at the optimum injection advance.

The selection of fuel flow rates over fuel-air ratios as a controlling parameter was connected to the method of controlling the inlet air temperature. The regulation of inlet air temperature could be performed in one of three ways.

The inlet air temperature after water injection could be held constant. This requires increasing the inlet air temperature prior to water injection due to the evaporative cooling of the inlet air charge by the water spray. This procedure reduces the inlet air flow and hence volumetric efficiency by an amount dependent upon the degree of vaporization of the water injected. Thus, with increasing water flow and hence increasing air temperatures prior to water injection and decreasing inlet air flow, maintaining a constant F/A ratio for a particular test run would be both difficult and time consuming. Therefore, a constant fuel flow rate is the better parameter to keep constant.

The inlet air temperature before water injection could be held constant. This would tend to increase inlet air flow and hence volumetric efficiency due to evaporative cooling by the water spray. In this case, as before, a constant fuel

flow rate is again the better parameter to control.

Finally, the inlet air temperature before and after injection, and possibly fuel flow rate, could be varied over a small given range to produce a constant F/A ratio for a particular test run. In as much as this procedure allows the energy input (fuel) to be varied, and the volumetric efficiency to be even more confusing, the author felt this procedure was not desirable even if feasible from a time standpoint.

The author selected the first method of maintaining a constant inlet air temperature after injection. This was primarily to eliminate the variation in one variable that effects the calculations, the actual inlet air temperature. It was hoped that more meaningful overall results would therefore be obtained. This inlet air temperature was held at 90F.

The other operating variables held constant were inlet cooling water to both the head and the barrel at 170 F and the oil temperature at 160 F.

(3) Procedure

Testing started at the lowest RPM, lowest fuel rate and lowest W/F ratio in the unsupercharged condition. As soon as the range of W/F ratios at this lowest fuel flow rate was covered, the next higher fuel flow rate was examined through the range of W/F ratios. Upon completing the range of fuel flow rates for the lowest speed, the next higher

speed was examined in the same manner. When the unsupercharged operation was complete, the supercharged conditions at 10" Hg were covered.

Friction readings were taken at the end of each day of operation or at the end of each speed run if the latter occurred before a day's operations were completed.

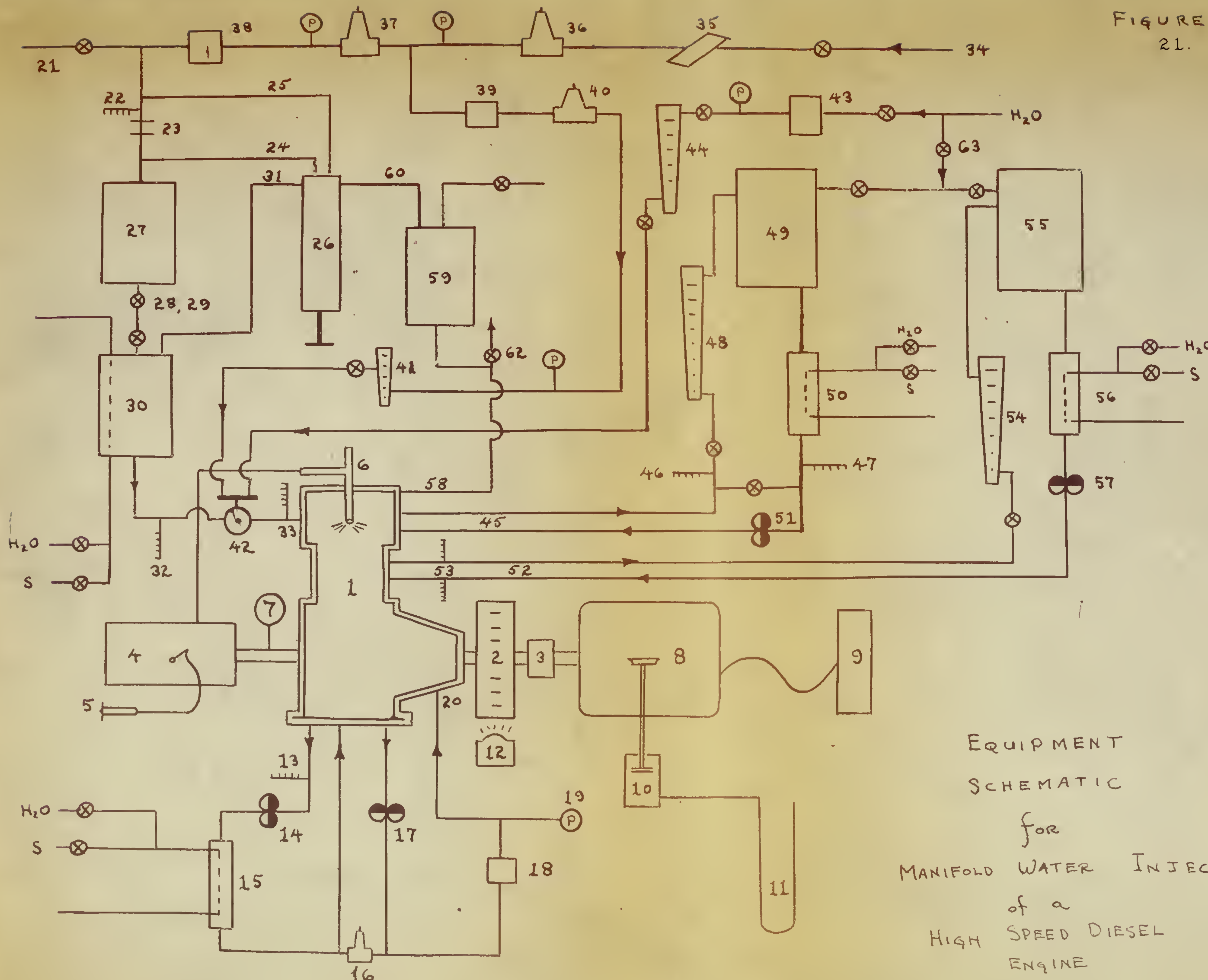
Sufficient time was allowed at each operating point to insure equilibrium of all conditions. This time of course varied with the degree of change in operating conditions but generally was at least 30 minutes. During this time continual checking and periodic correcting of all conditions helped insure steady state operation at the time the data on the operating point was recorded.

Indicator diagrams were not made due to the inability of one operator to handle the engine and all controls as well as the indicator diagram apparatus.

F. EQUIPMENT SCHEMATIC

A complete equipment schematic appears in Figure II-3 with a legend in Table II - 3.

Figure II 3



Equipment Schematic for Manifold Water Injection of a High Speed Diesel Engine

TABLE II -3

LEGEND TO FIGURE II - 3

No.	Description
1.	engine
2.	flywheel
3.	speed reducer, timing belt type, 2:1 ratio
4.	fuel injection pump, timing adjustment and flow instrumentation
5.	main fuel delivery control
6.	fuel injection nozzle in cylinder head
7.	tachometer
8.	electric dynamometer
9.	dynamometer controls
10.	brake load hydraulic cylinder
11.	manometer, brake load read-out
12.	line frequency stroboscope aimed at flywheel
13.	thermocouple, crankcase oil temperature
14.	circulating pumper for lube oil
15.	heat exchanger for lube oil
16.	pressure relief valve for lube oil
17.	pressure pump for lube oil
18.	oil filter
19.	oil pressure gauge
20.	pressurized lube oil delivery to bearings
21.	engine air intake at atmospheric pressure
22.	thermometer, air before metering orifice

23. air metering orifice, ASME square edge type

24. manometer line, ΔP across air metering orifice

25. manometer line, air before metering orifice

26. manometers

27. air surge tank

28. air throttle valve

29. emergency air shut off valve, quick-acting type

30. tank for heating or cooling inlet air

31. manometer line, inlet air pressure

32. thermocouple, inlet air temperature before water spray

33. thermocouple, inlet air temperature after water spray

34. compressed air supply for supercharging

35. strainer, compressed air

36. regulator, reduces lab mains to 50 psi

37. regulator, controls supercharging pressure

38. supercharging air filter

39. spray valve air filter and water separator

40. regulator, controls spray valve air pressure

41 rotometer, spray valve compressed air

42. water spray valve mounted in intake manifold

43. spray water filter

44. rotometer, spray water supply

45. head cooling water lines in and out of engine

46. thermocouple, shunt inlet

47. thermocouple, shunt outlet

48. rotometer, shunt circuit

49. expansion tank for head and shunt cooling circuits

50. heat exchanger, shunt circuit

51. pump for head and shunt cooling circuits

52. barrel cooling water lines in and out of engine

53. thermocouples, barrel inlet and outlet

54. rotometer, barrel cooling circuit

55. expansion tank for barrel cooling circuit

56. heat exchanger, barrel cooling circuit

57. pump for barrel cooling circuit

58. exhaust manifold

59. exhaust cooling and surge tank

60, manometer line, exhaust pressure

61. exhaust throttle valve and discharge to trench

62. valve to allow exhaust discharge into test cell for observation

63. cooling water surge tank refill lines

CHAPTER III

EXPERIMENTAL RESULTS

A. INTRODUCTION

This chapter presents the results of the investigation on intake manifold water injection in the form of graphs of the characteristics investigated versus the water-fuel (W/F) ratio. Not every operating point investigated is presented as the author selected the most representative cases for illustration. Although generally the data scatter was small and the results very satisfactory, inclusion of all curves for all test runs would be somewhat repetitive.

Surprisingly, the engine ran well with water injection up to values equal to the fuel flow rate (W/F of 1.0). Never did the engine lope along disliking the water injection and it remained as smooth running with the injection as without.

B. HEAT REJECTION

The results on the reduction of heat rejection were extremely satisfactory and all RPM's and fuel flow rates are shown in the curves as this characteristic was the primary reason for the investigation.

As implied in the apparatus text and shown clearly on the equipment schematic, the water cooling system was divided into a head and a barrel circuit. As the heat rejected by the head is more clearly indicative of the combustion process

and more sensitive to changes in operating parameters, this system is more representative of the internal cooling effect of the water. As can be seen from the following curves of $\dot{Q}_{head}$ versus W/F ratio a definite decrease in the amount of heat rejected occurs with increasing W/F ratio. The effect appears more pronounced at the richer F/A ratios to the higher fuel flow rates. A comparison of the unsupercharged operation versus the supercharged operation indicates that the degree of reduction in heat rejection appears to be more a function of the F/A ratio than the fuel flow rate. More reduction in heat rejection seems to occur in the unsupercharged condition.

To illustrate the effect of combined head and barrel circuits, two curves of $\dot{Q}_{total}$ versus W/F are included. The author found that the barrel heat rejection was insensitive to water injection and remained fairly constant for a given F/A ratio and speed. There was considerable scatter of data in this area however as the barrel cooling circuit is not "shunted" as is the head circuit and thus accurate heat transfer data is more difficult to obtain due to the small change in temperature in cooling water measured across the barrel.

C. INDICATED EFFICIENCY

Indicated efficiency is a convenient tool for comparison between the various operational points as it eliminates, or smooths out, the small uncontrollable changes in operating

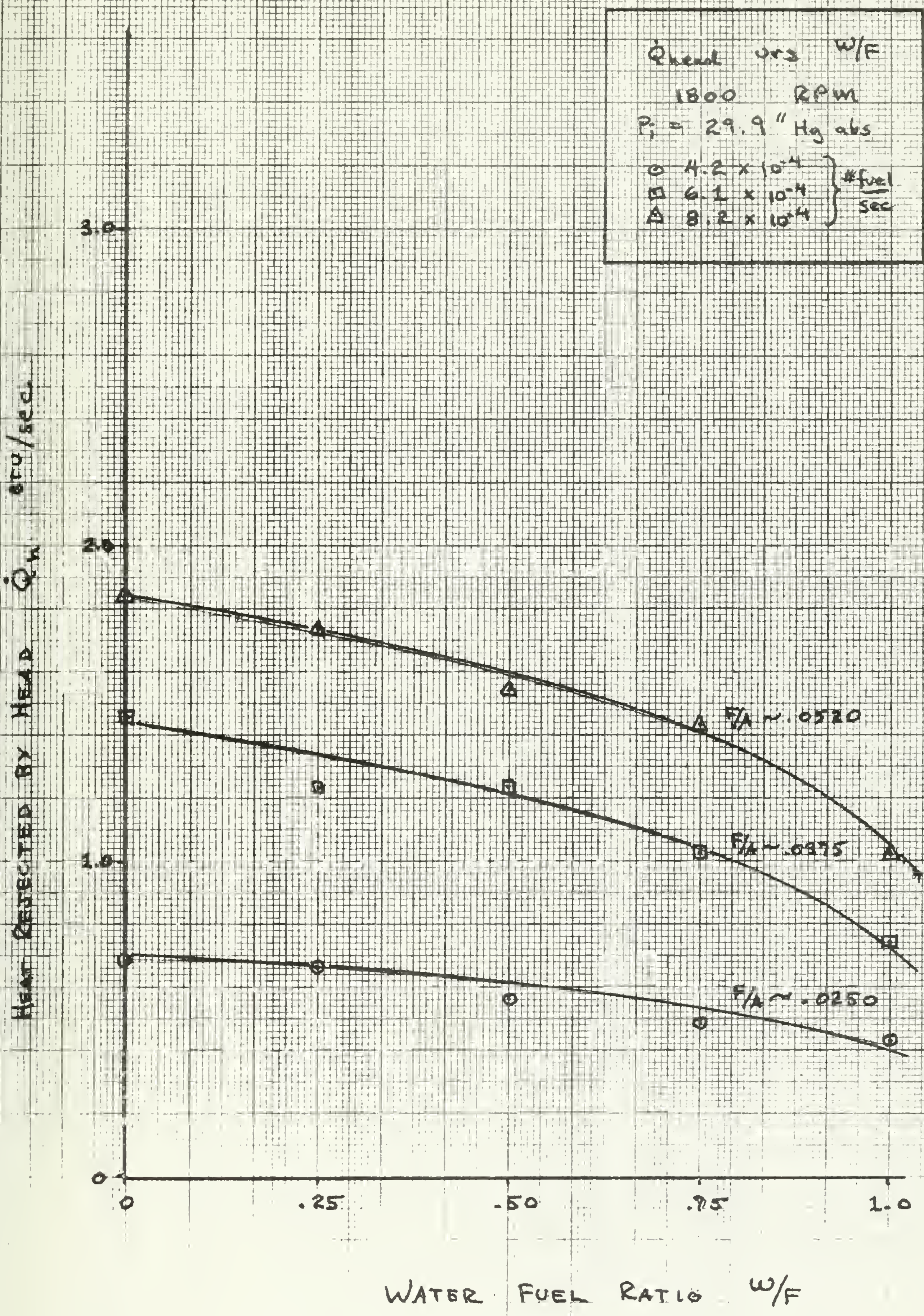

$\dot{Q}_{head}$ vrs W/F
1800 RPM
P_i = 29.9" Hg abs
⊙ 4.2 x 10^{-4}
⊡ 6.1 x 10^{-4}
△ 8.2 x 10^{-4}
#fuel/sec
F/A ~ .0520
F/A ~ .0375
F/A ~ .0250
HEAT REJECTED BY HEAD $\dot{Q}_h$ BTU/SEC
0
1.0
2.0
3.0
0
.25
.50
.75
1.0
WATER FUEL RATIO W/F

FIGURE III- 1B

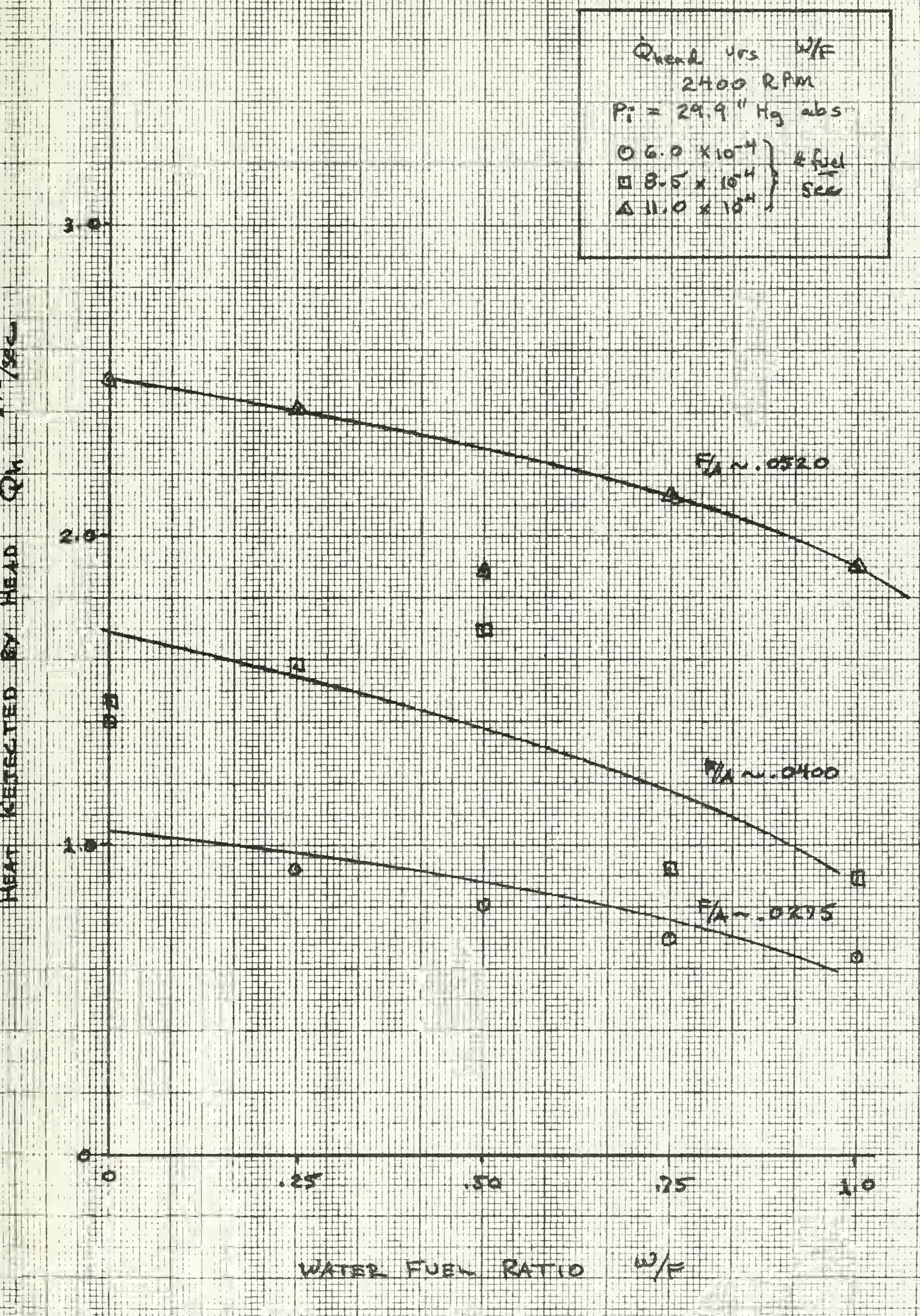
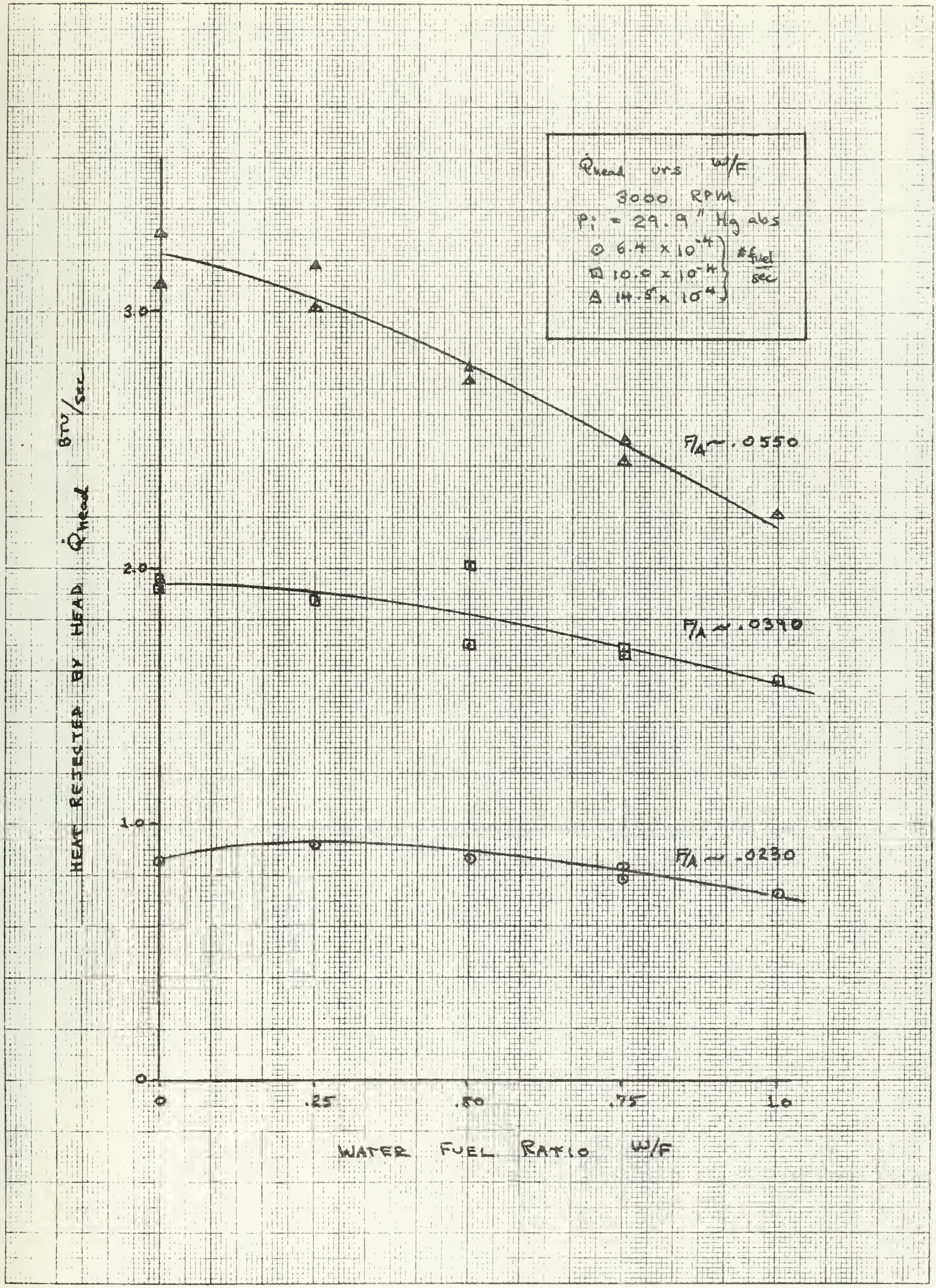


FIGURE III - 1C



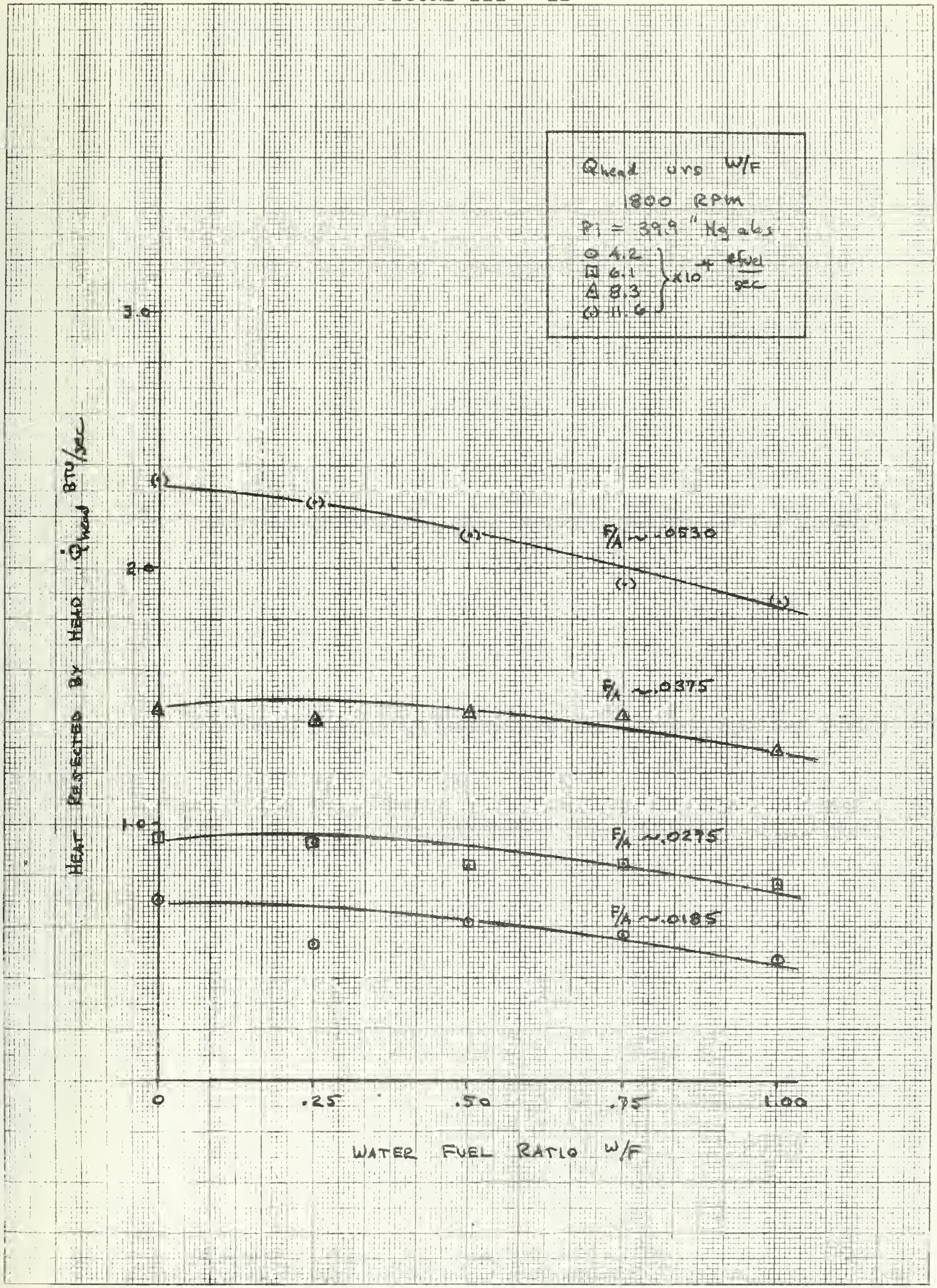

Qhead vrs W/F
1800 RPM
P1 = 39.9 " Hg abs
4.2
6.1
8.3
11.6
x10^-4 #fuel/sec
3.0
2.0
1.0
F/A ~ .0530
F/A ~ .0375
F/A ~ .0275
F/A ~ .0185
HEAT REJECTED BY HEAD Qhead BTU/sec
0
.25
.50
.75
1.00
WATER FUEL RATIO W/F

FIGURE III - 1E

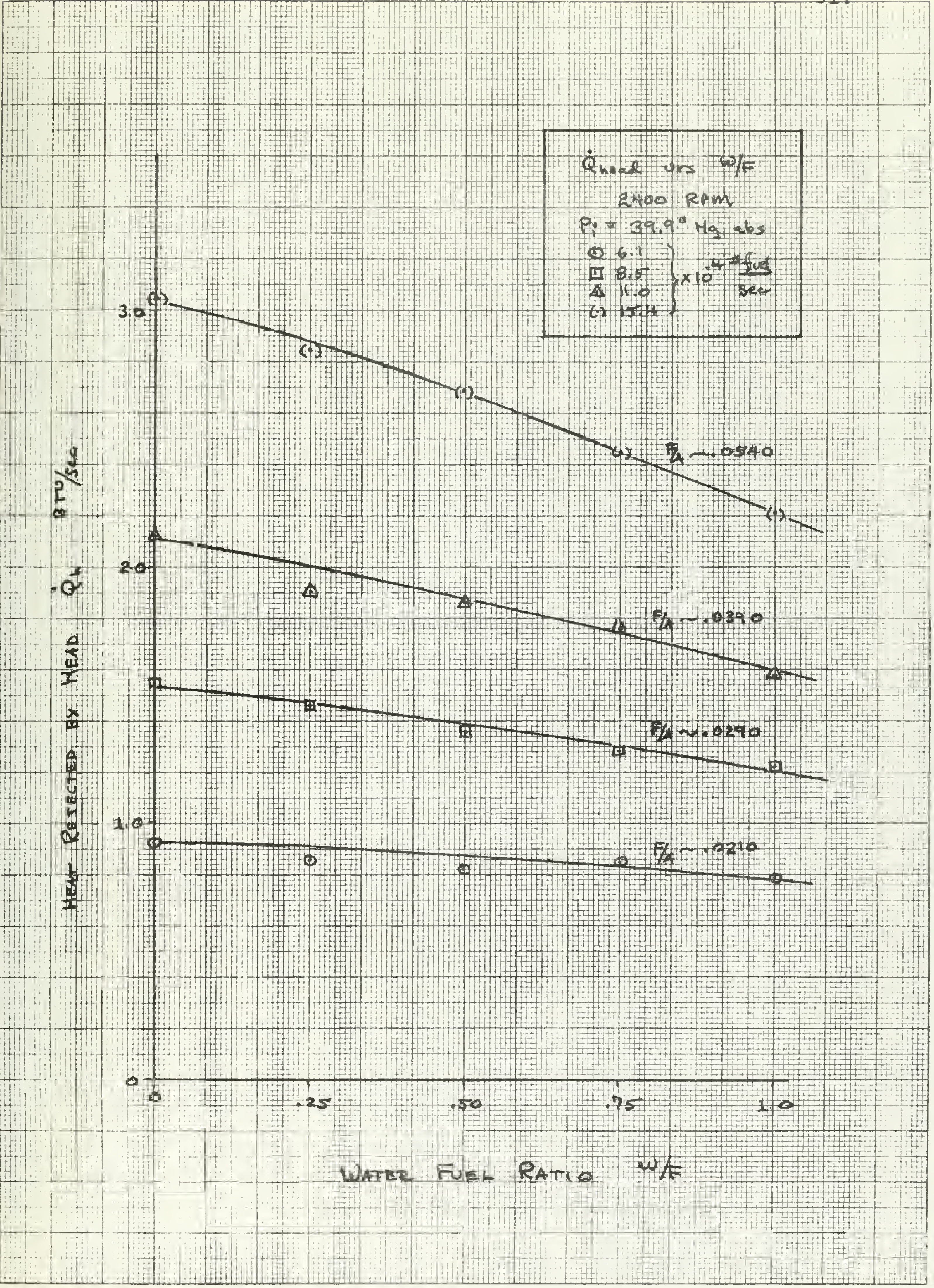


FIGURE III -1F

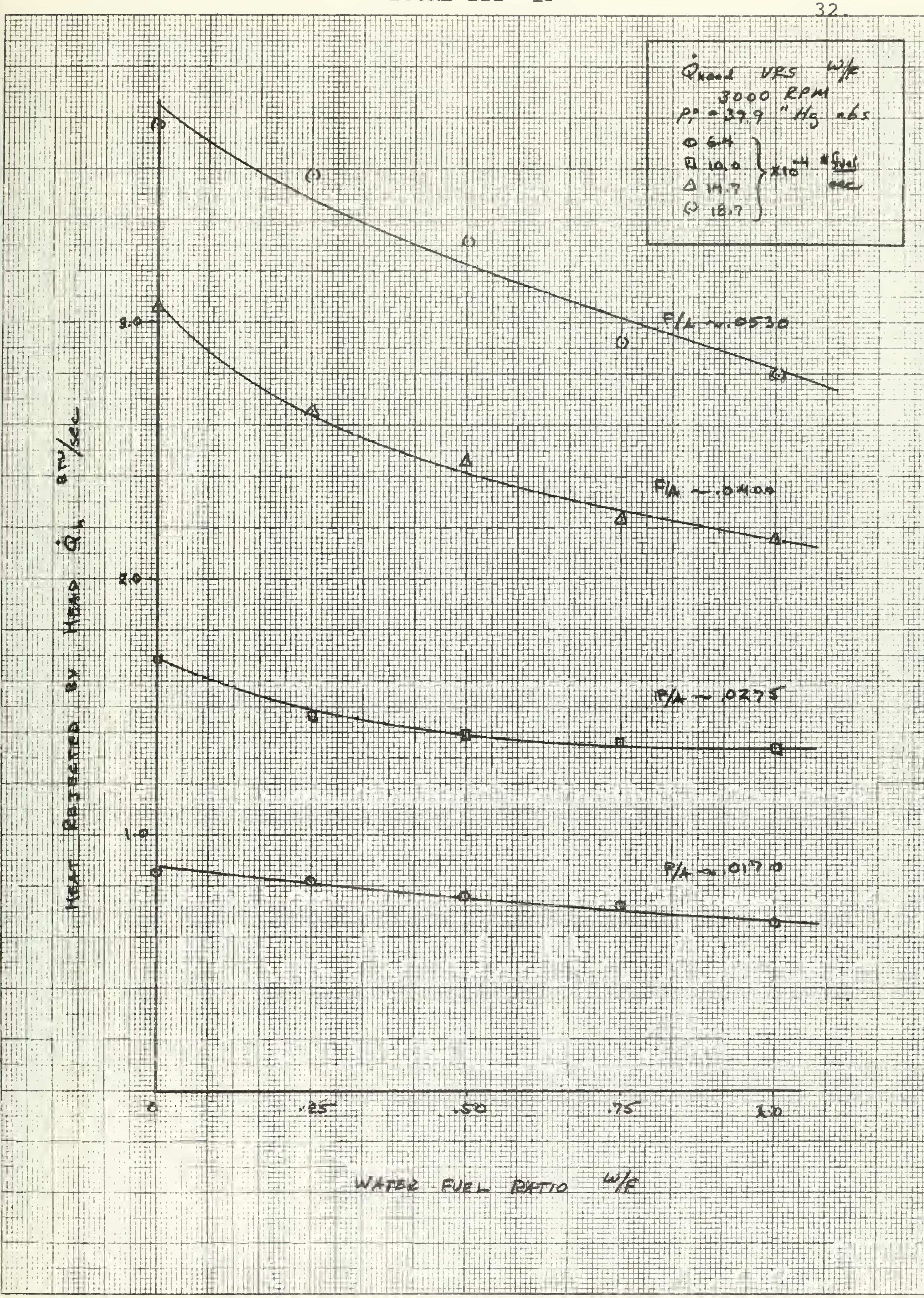


FIGURE III - 2A

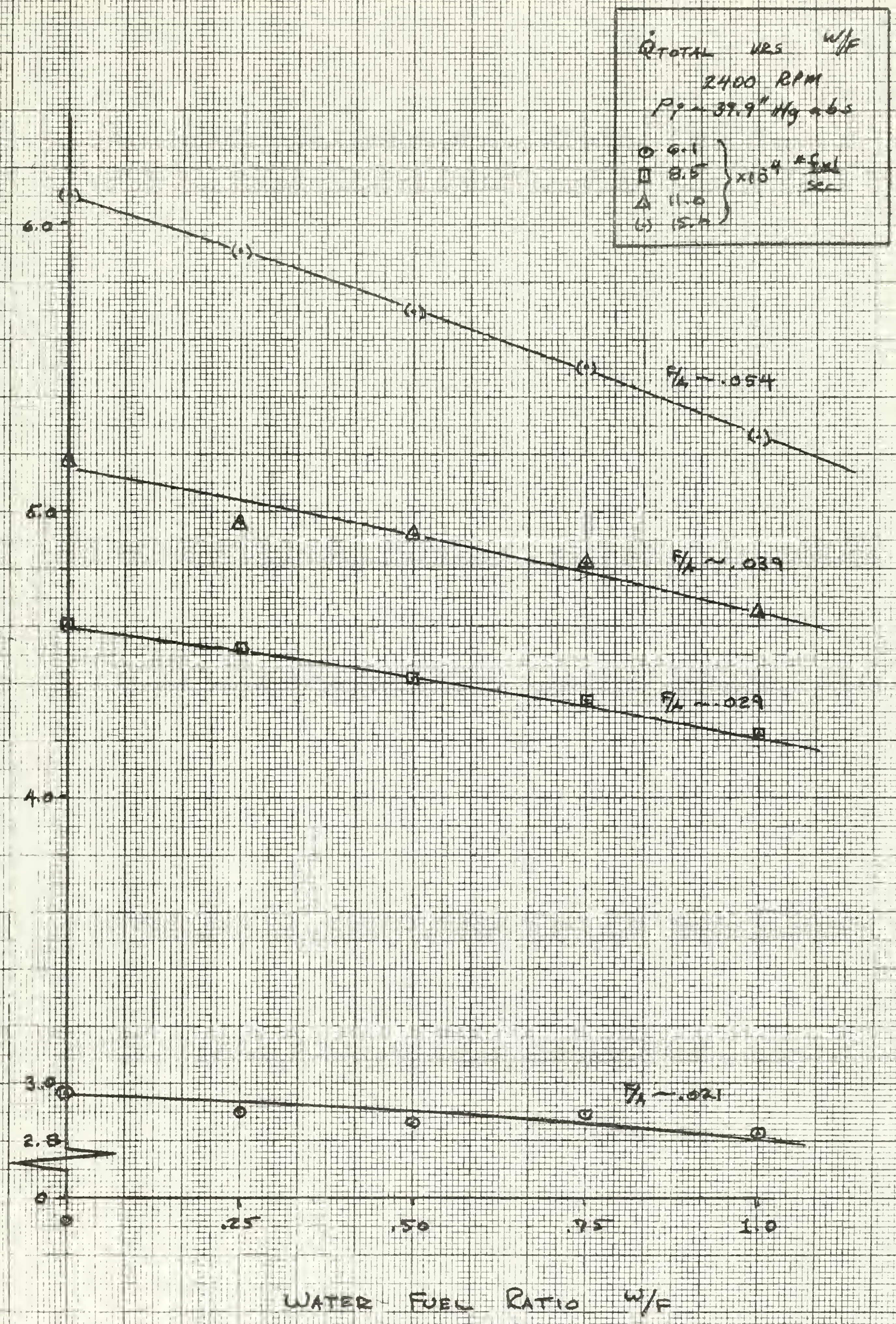
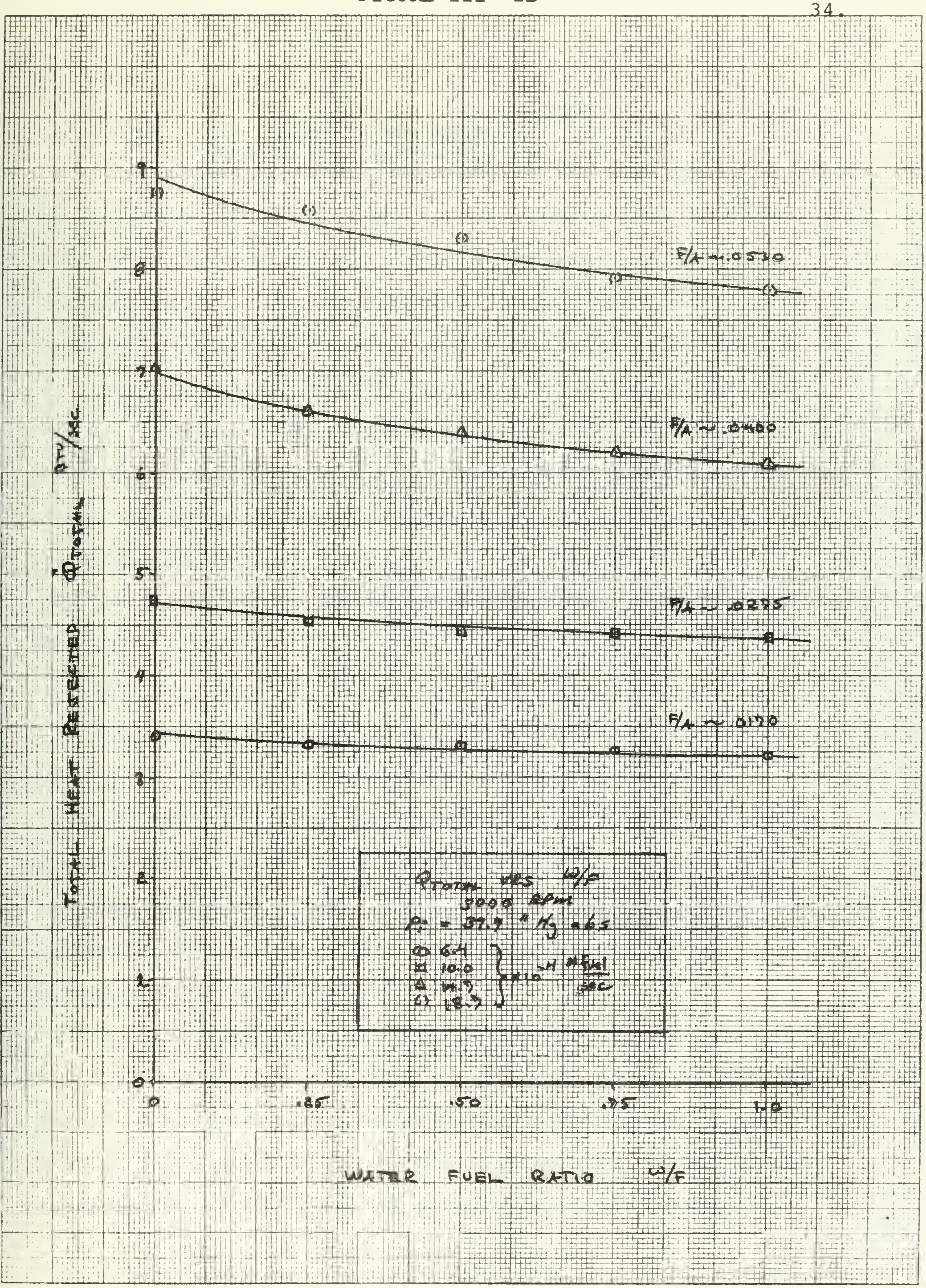


FIGURE III -2B



conditions that so often show up as rather noticeable effects in other characteristics. For example, a slight change in fuel flow rate would have a significant effect on power output but a rather small change in indicated efficiency. Thus, indicated efficiency appears to be an excellent indication of how 'well' the engine likes water injection.

As can be seen from the curves of indicated efficiency versus W/F, the efficiency decreases somewhat, perhaps several percentage points, but there are no drastic decreases. Comparison between the unsupercharged operation and the supercharged operation illustrates a more marked decrease for the former while the latter decreases only slightly if al all.

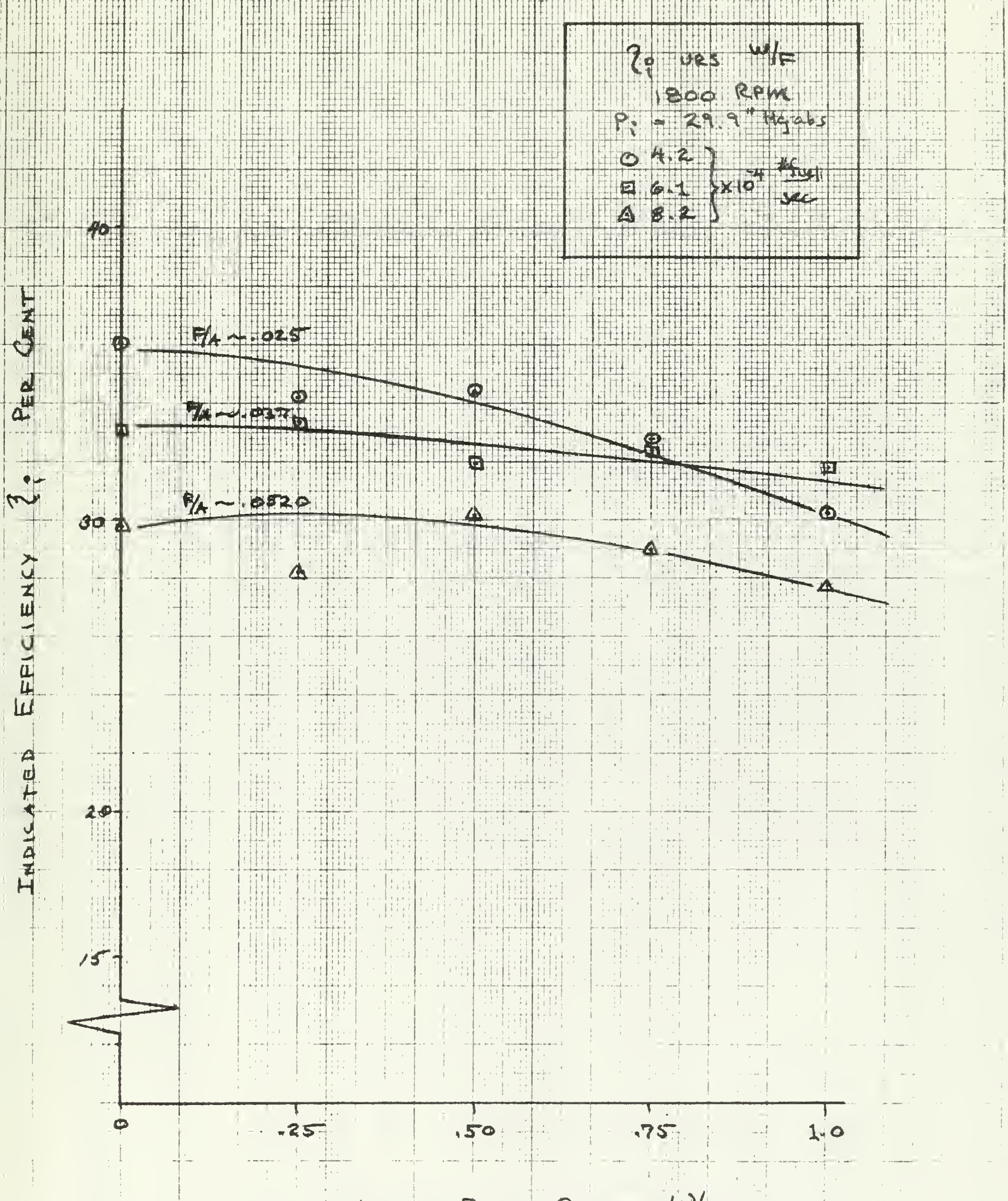
D. IMEP

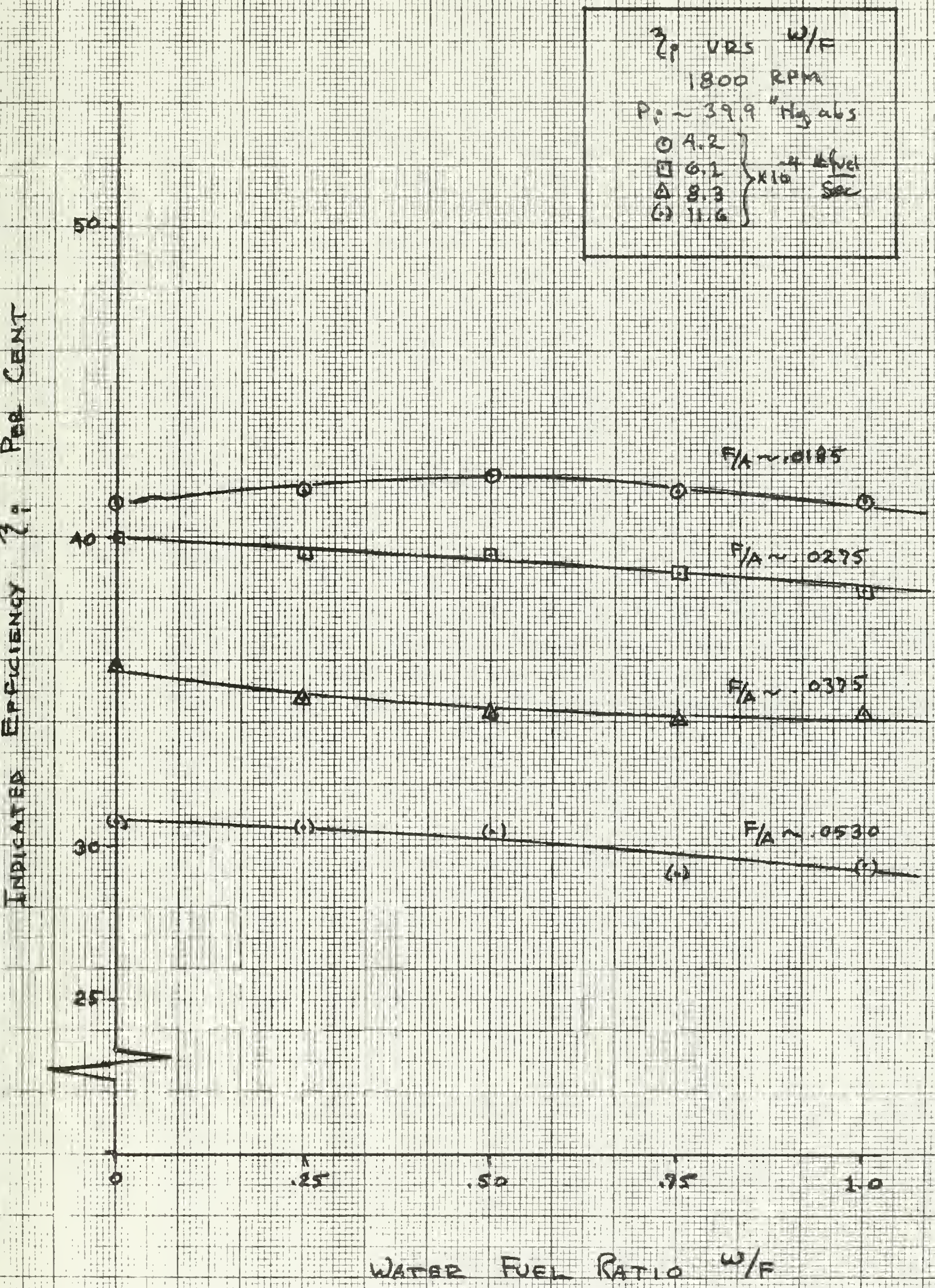
The effect of water injection on the IMEP is rather small, but as with the indicated efficiency, the IMEP tends to decrease. The curves illustrated were selected for minimal variation in the fuel flow rate over the test run in order that the trends of the power decrease would be more representative of the effect of only increased W/F ratio.

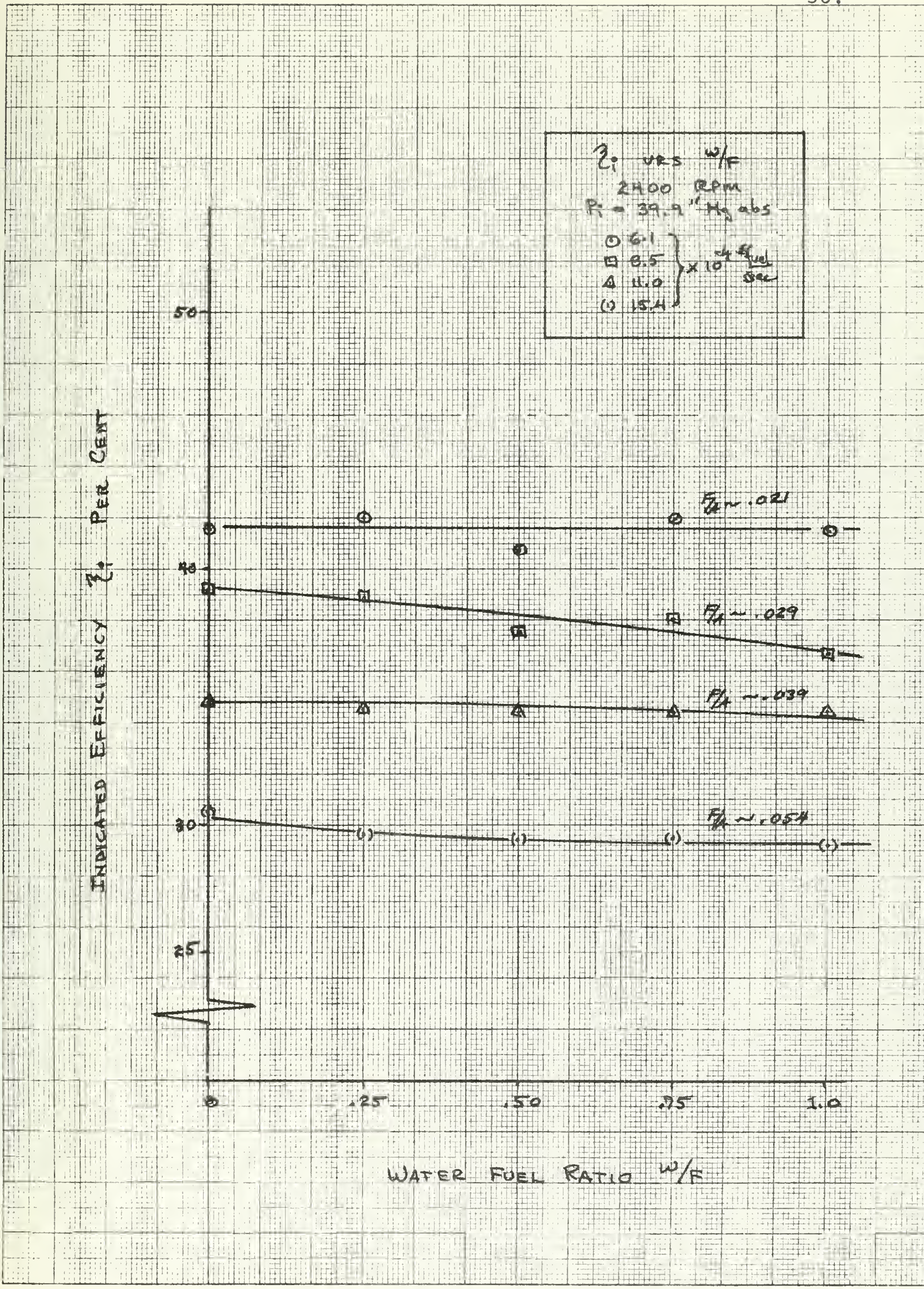
E. VOLUMETRIC EFFICIENCY

As discussed under the TEST PROCEDURE section of Chapter II, the volumetric efficiency changes with increased W/F ratio become considerably more complex than with changes in normal engine operation. This, as mentioned earlier, is due to the vaporization of the water spray which manifests itself in

FIGURE III - 3A




η_i VRS W/F
1800 RPM
$P_i \sim 39.9$ "Hg abs
⊙ 4.2
□ 6.1
△ 8.3
(·) 11.6
$\times 10^{-4}$ # fuel / Sec
F/A ~ .0185
F/A ~ .0275
F/A ~ .0375
F/A ~ .0530
50
40
30
25
0
.25
.50
.75
1.0
Indicated Efficiency η_i Per Cent
Water Fuel Ratio W/F



η_i vrs W/F
2400 RPM
P_i = 39.9" Hg abs
6.1
8.5
11.0
15.4
$\times 10^{-4}$ #fuel/sec
F/A ~ .021
F/A ~ .029
F/A ~ .039
F/A ~ .054
50
40
30
25
0
.25
.50
.75
1.0
Indicated Efficiency η_i Per Cent
Water Fuel Ratio W/F

FIGURE III -3D

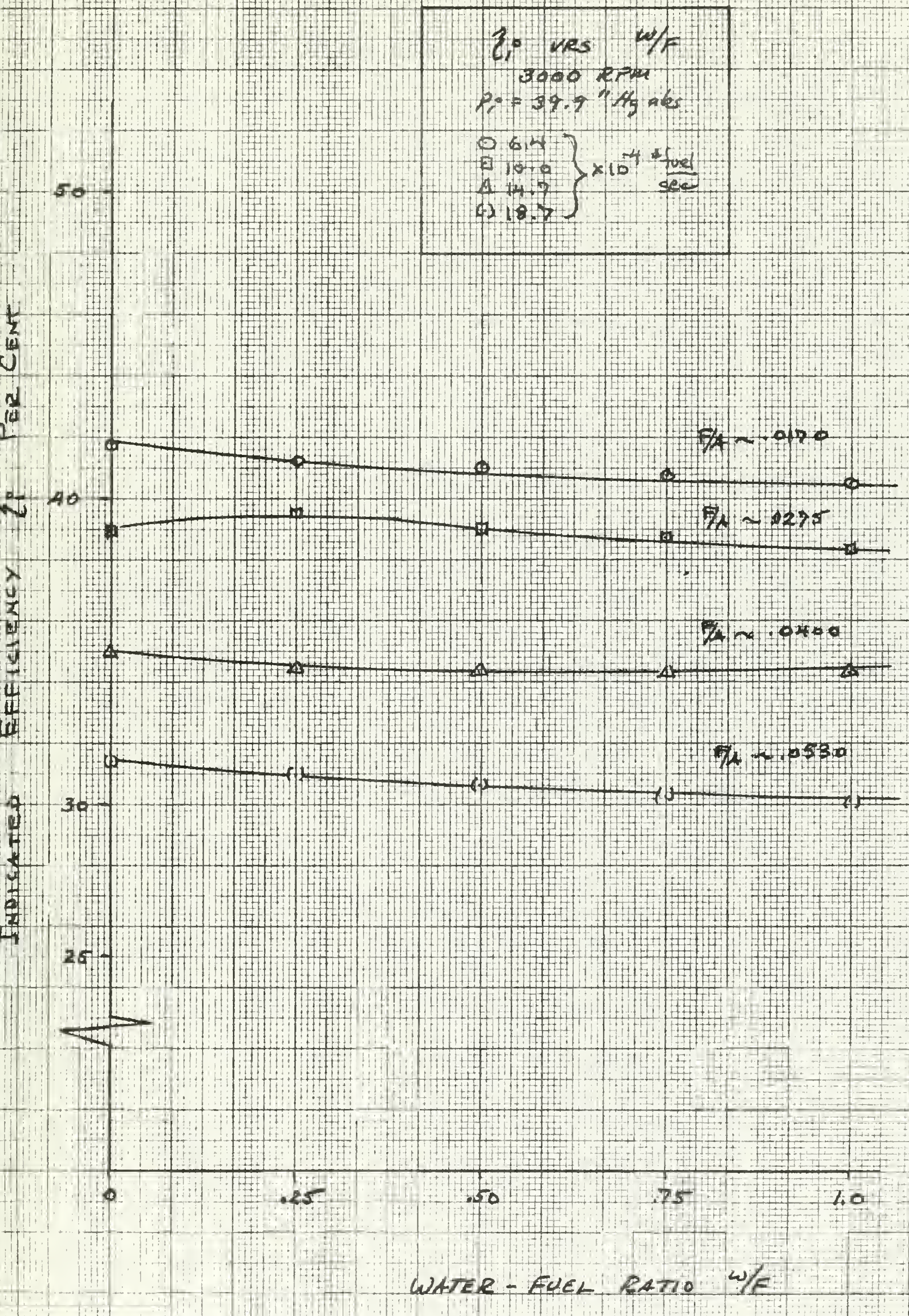


FIGURE III -4A

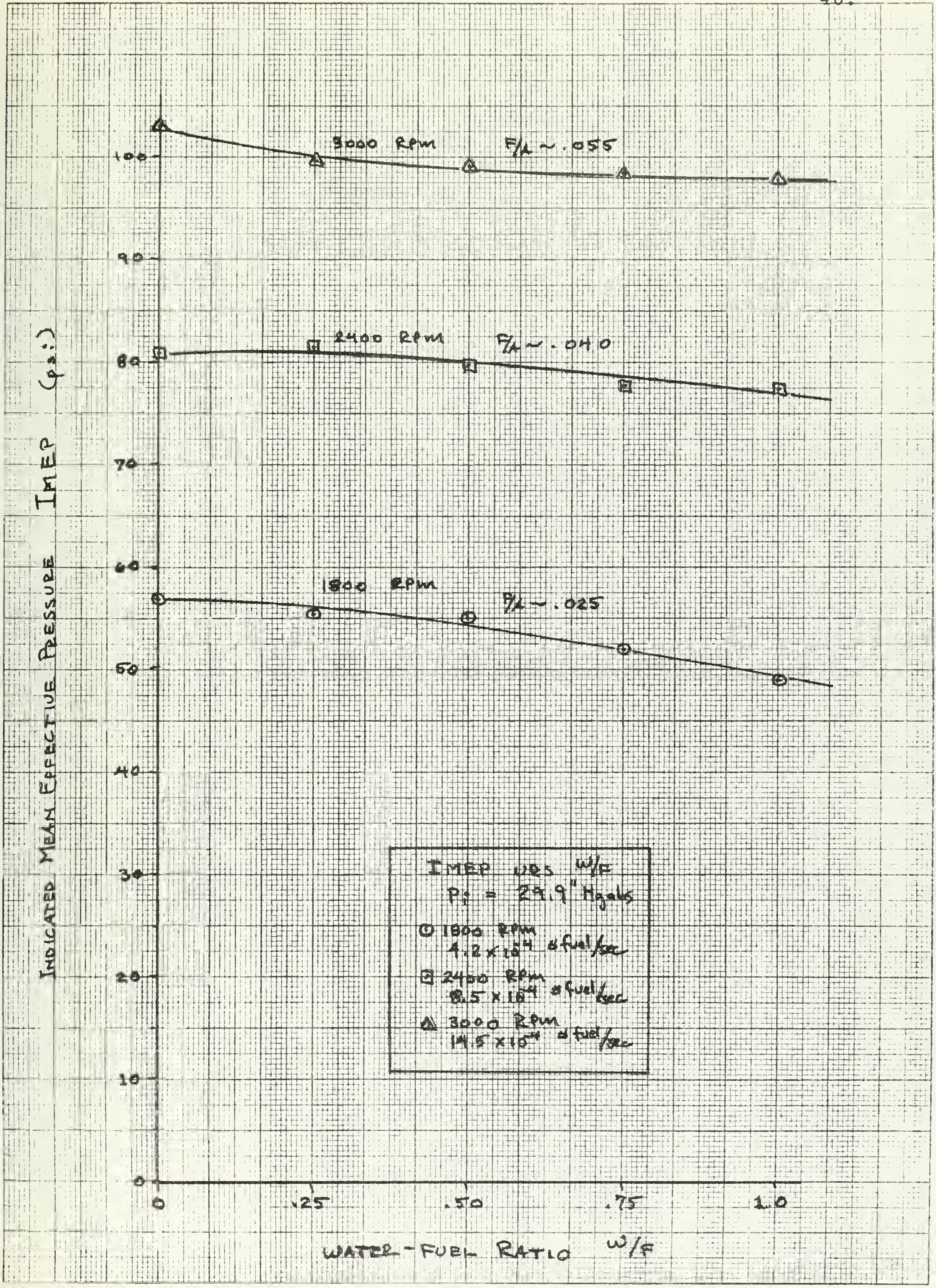
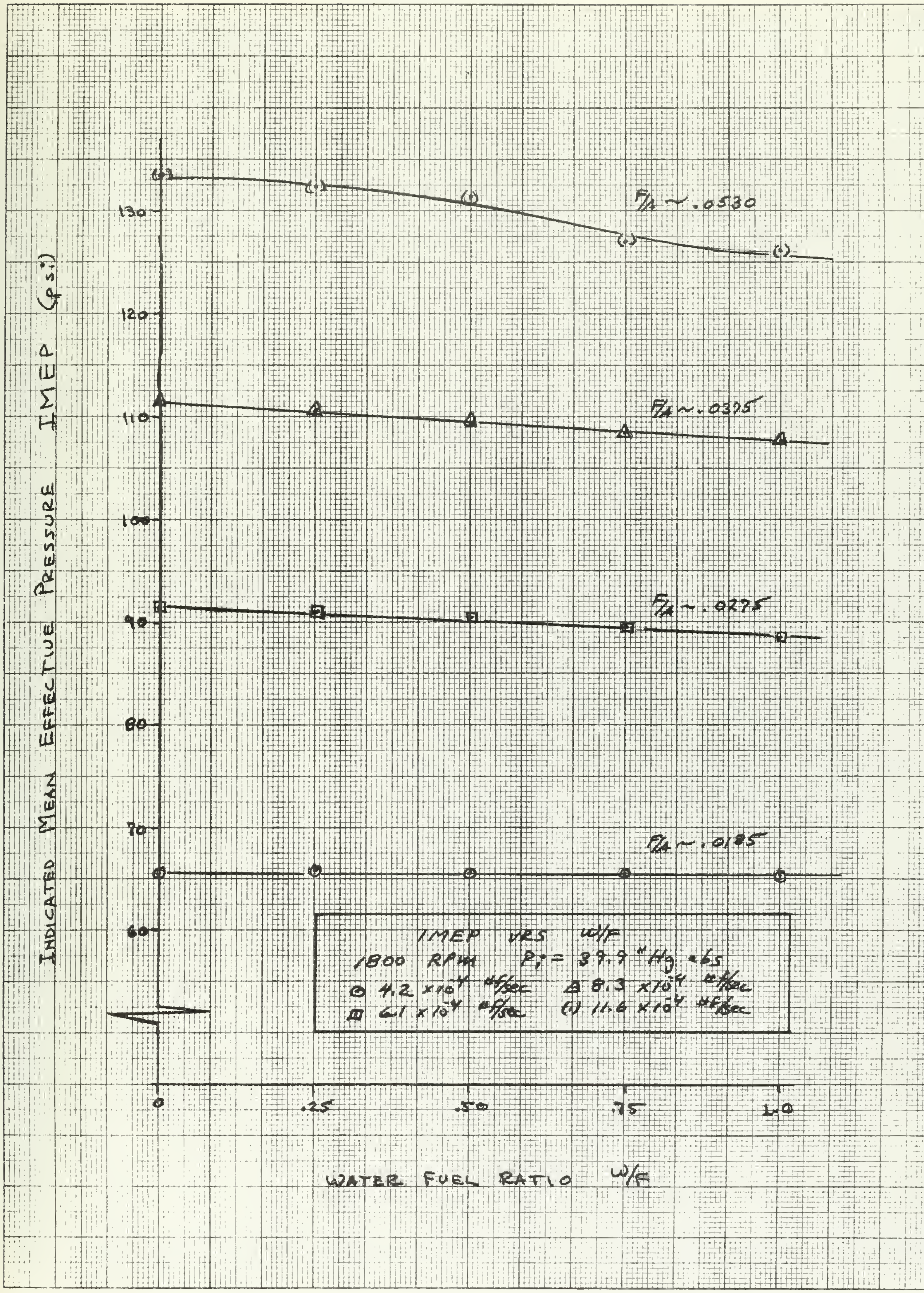
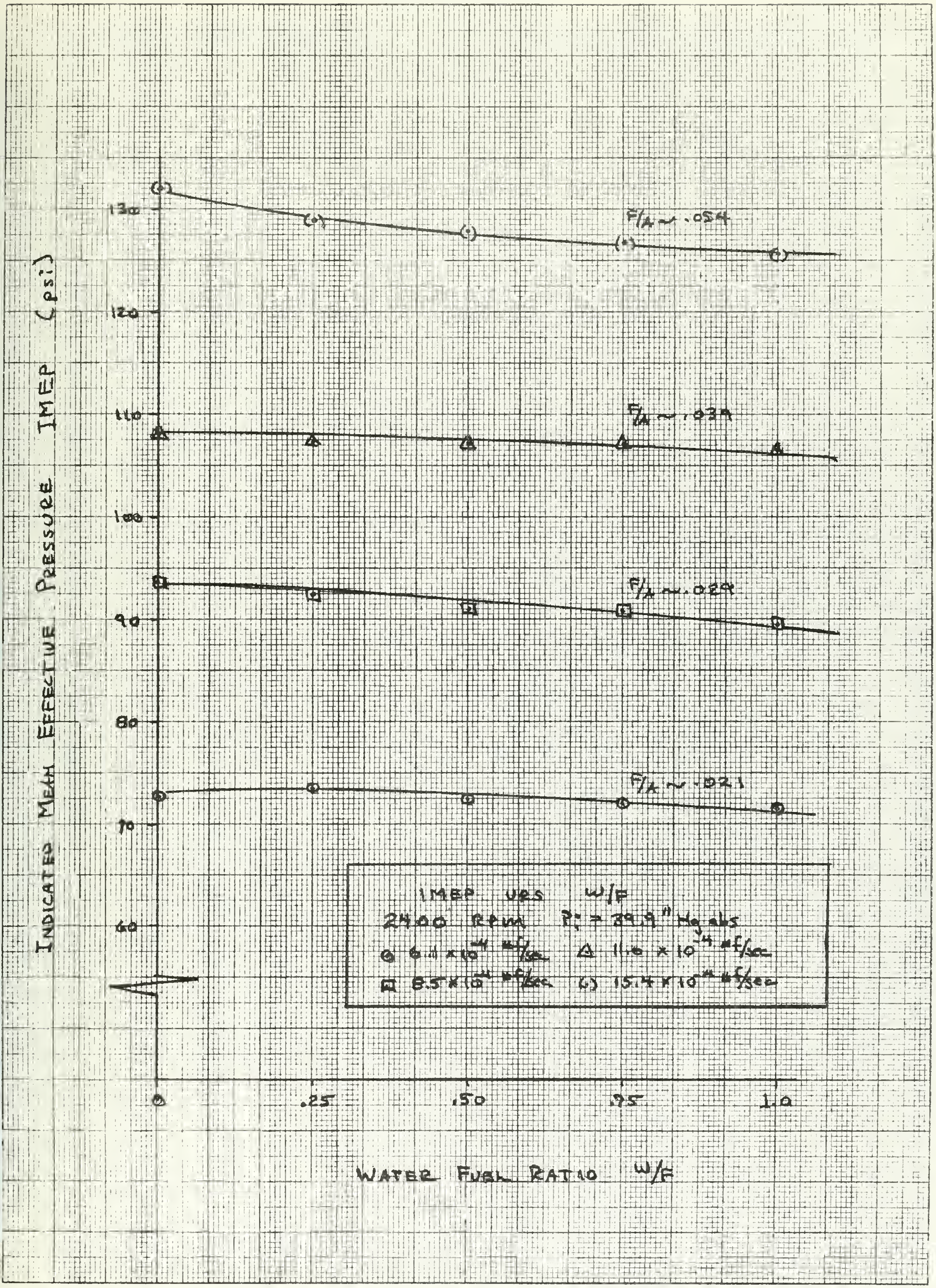


FIGURE III -4B





Indicated Mean Effective Pressure IMEP (psi)
130
120
110
100
90
80
70
60
F/A ~ .054
F/A ~ .039
F/A ~ .029
F/A ~ .021
IMEP vrs W/F
2400 RPM Pi = 39.9" Hg abs
6.1 x 10^-4 #f/sec
11.6 x 10^-4 #f/sec
8.5 x 10^-4 #f/sec
15.4 x 10^-4 #f/sec
0
.25
.50
.75
1.0
Water Fuel Ratio W/F

FIGURE III -4D

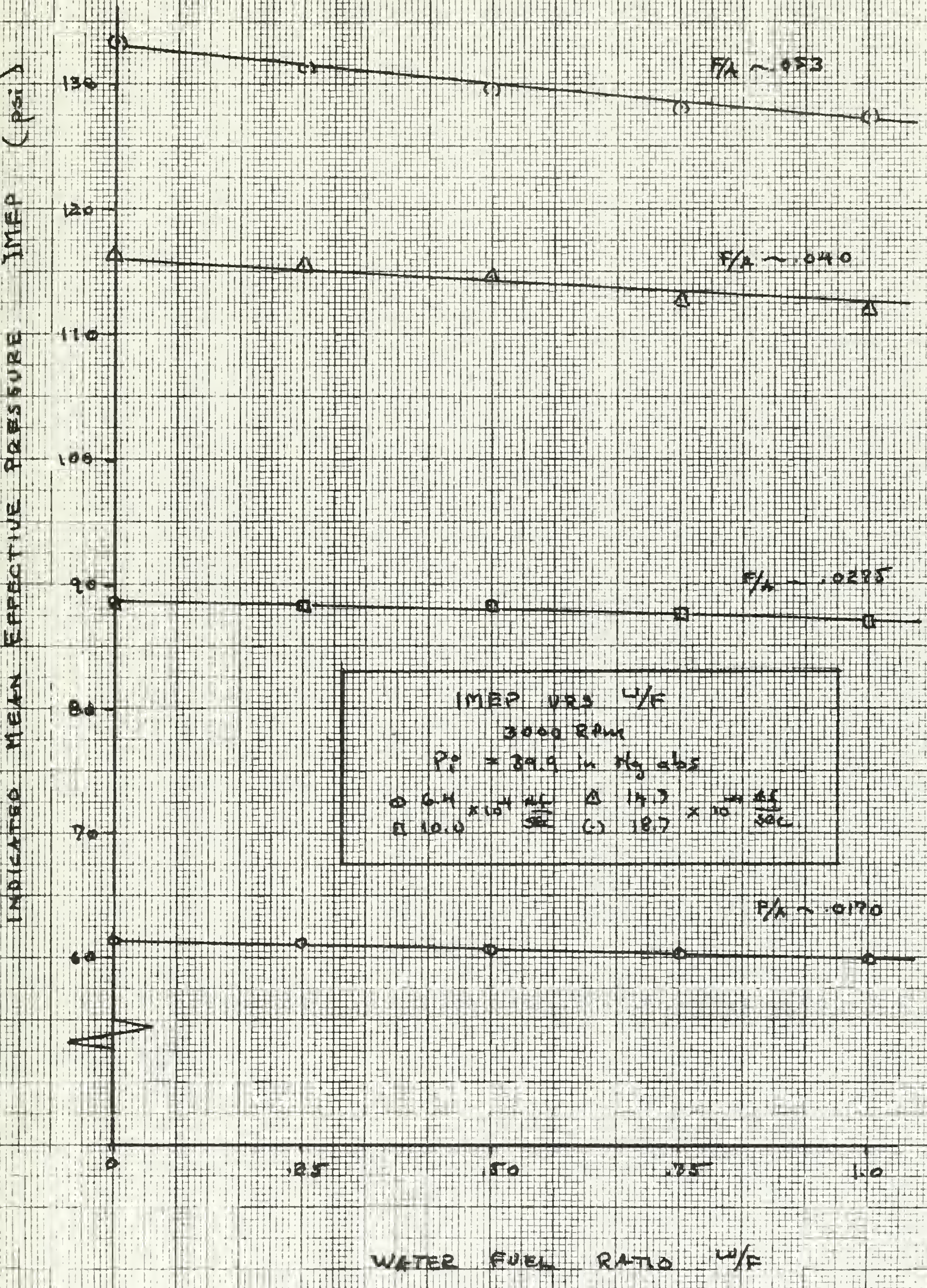


FIGURE III -5A

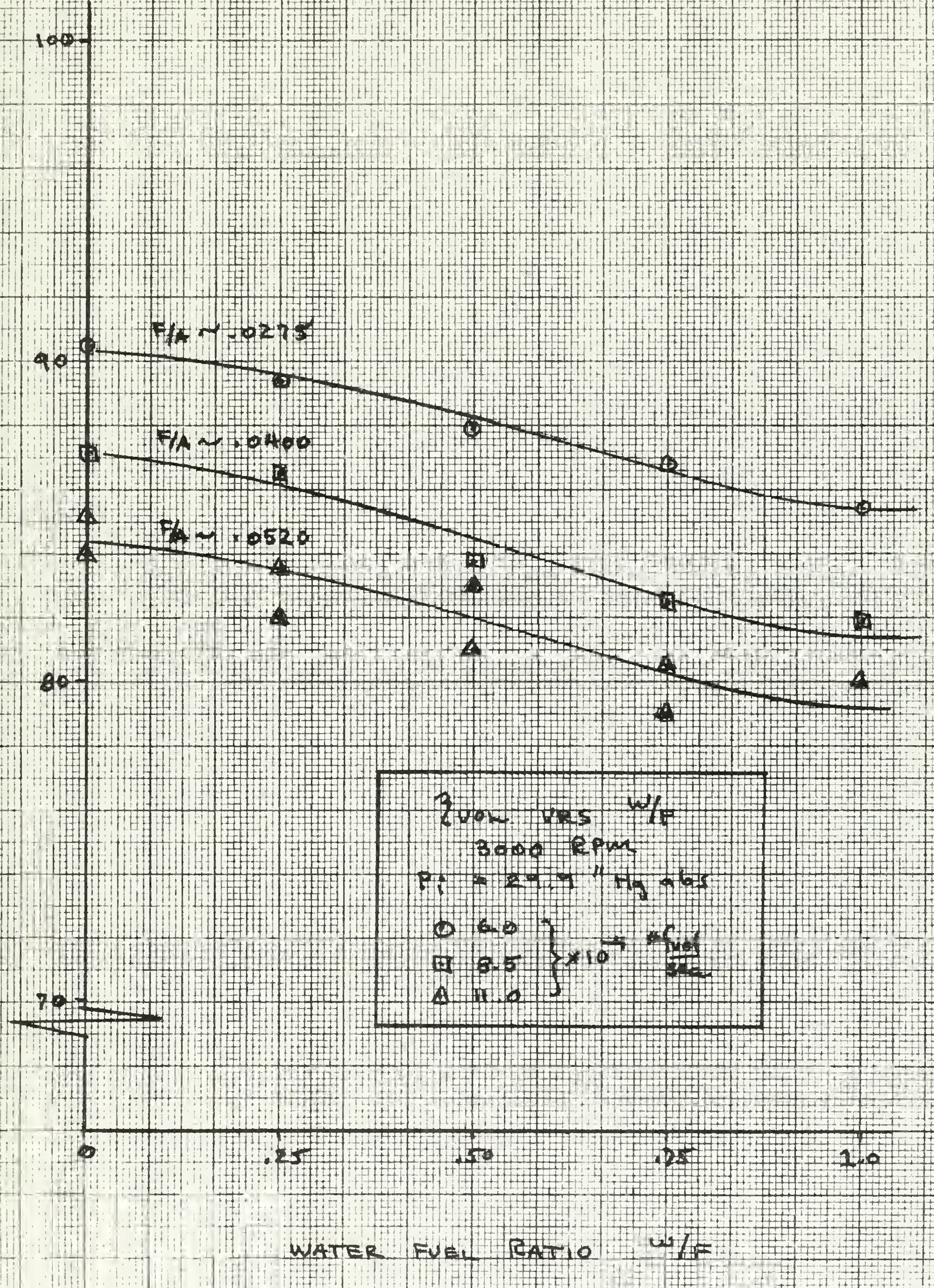
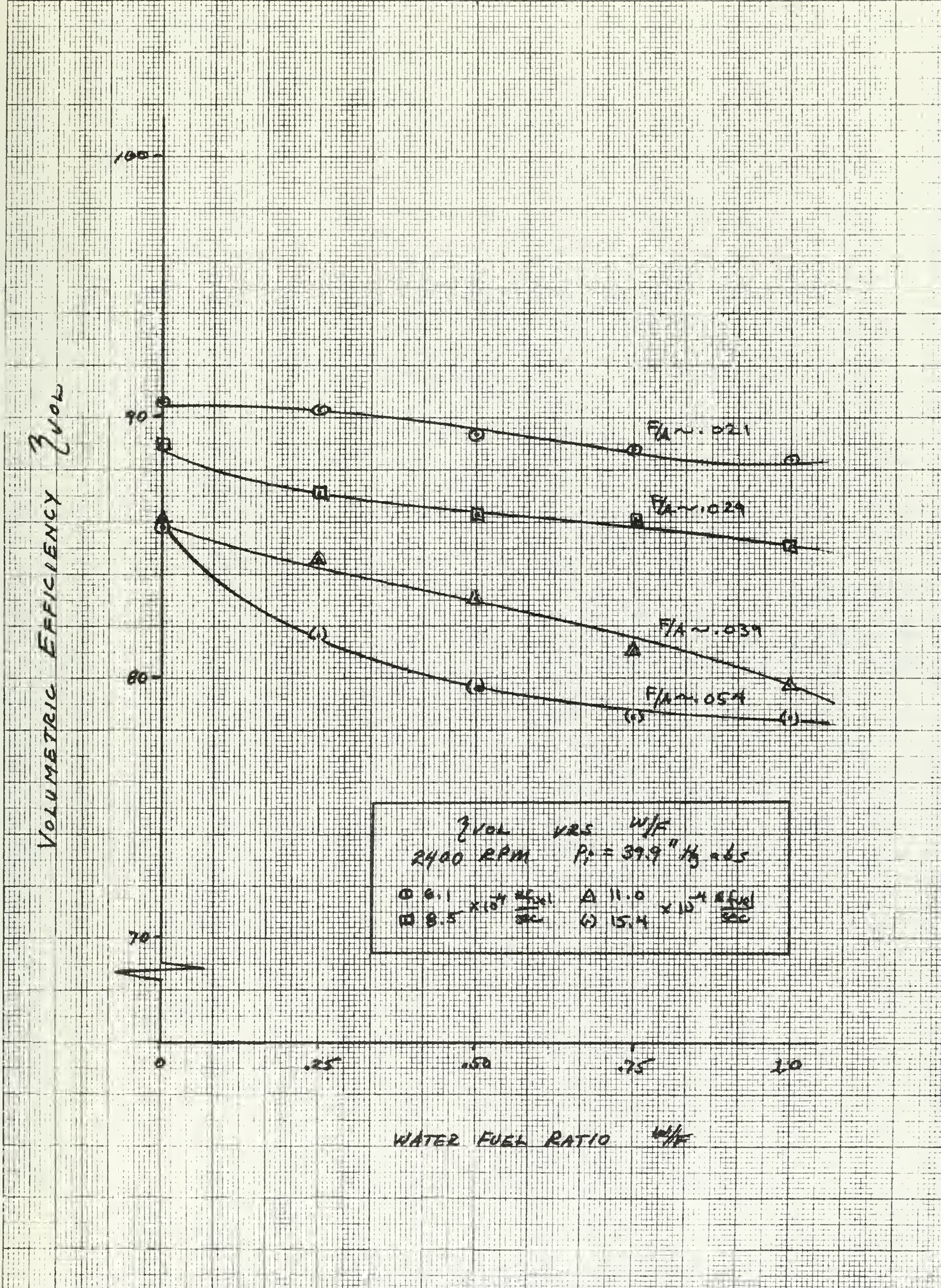
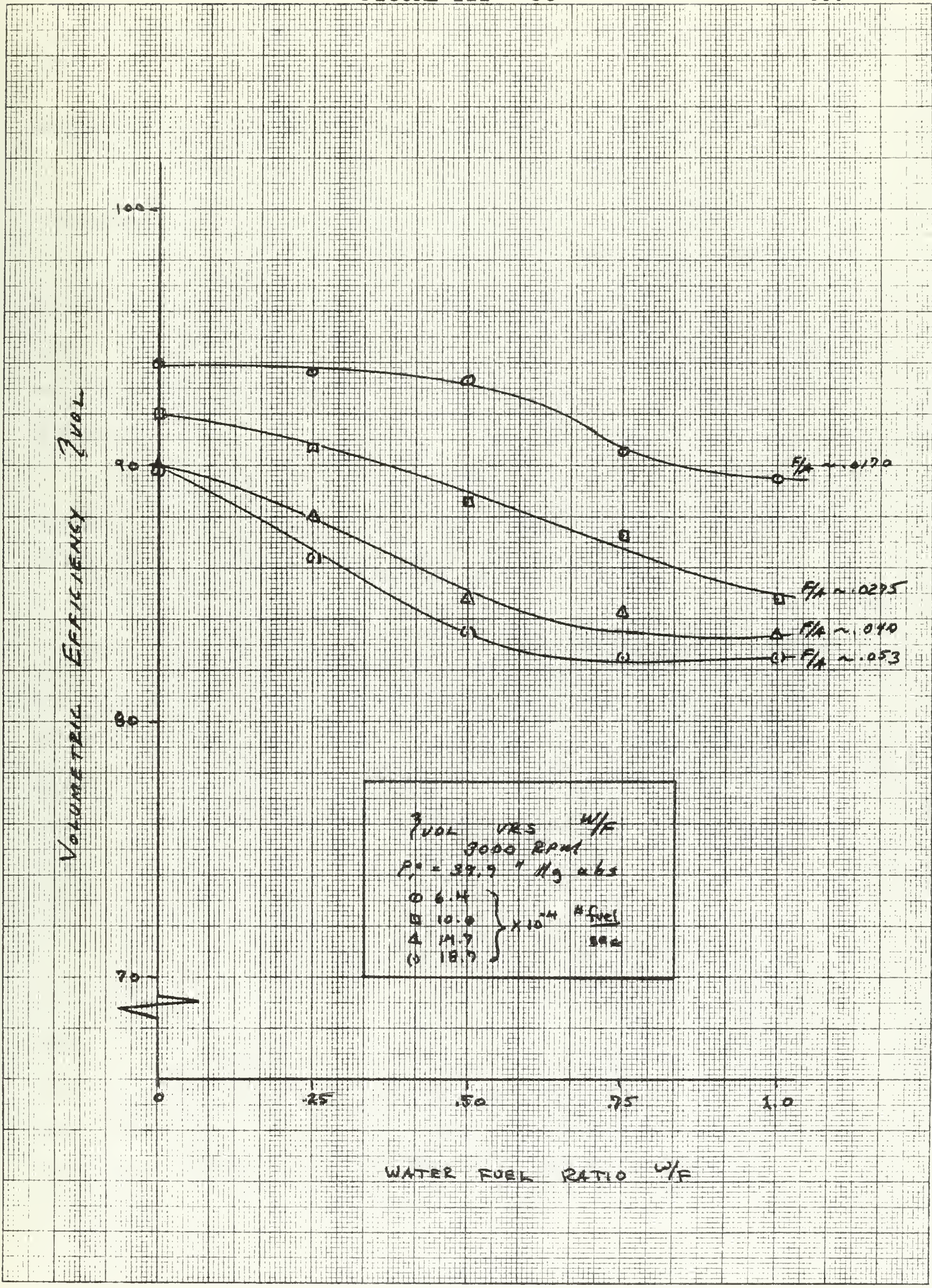


FIGURE III - 5B




100
90
80
70
VOLUMETRIC EFFICIENCY η_{VOL}
F/A ~ .0170
F/A ~ .0275
F/A ~ .040
F/A ~ .053
η_{VOL} VRS W/F
3000 RPM
P_1 = 39.9 " Hg abs
6.4
14.7
18.7
$\times 10^{-4}$ # fuel / sec
0
.25
.50
.75
1.0
WATER FUEL RATIO W/F

several ways, including evaporative cooling and dilution of the air charge by water vapor. The effect of engine operation on volumetric efficiency is therefore impossible to sort out. There is no way of knowing exactly how much of the water vapor is vaporized at each W/F ratio, nor is it certain that the actual intake temperature is truly represented by the thermocouple reading due to possible water droplets on the thermocouple.

Nevertheless, the data has been reduced to illustrate volumetric efficiency versus W/F. As expected, the volumetric efficiency decreases with increasing W/F ratio.

F. COMBUSTION CHARACTERISTICS

Even if indicator diagrams were taken during the course of of this investigation, determining the effect of water injection on the combustion would be difficult. Since indicator diagrams were not taken the problem appears insurmountable. There are,however, several reliable elements that help overcome this difficulty.

(1) Injection Advance

The results of the experimentation showed that in the unsupercharged operation optimum injection usually advanced about 5° between the W/F of 0 and 0.5. On several occasions this advance occurred at higher W/F ratios but no pattern was discernable. For the supercharged operation the optimum injection advance did not change at all for the range of W/F ratios.

(2) Smoke and Efficiency

Smoke is a reliable visual indication of the completeness of combustion in a diesel engine and a smokey exhaust generally means poor efficiency. There was no evidence that the water injection helps clean the smoke, in fact in the unsupercharged operation an increase in W/F usually caused the exhaust to smoke up a little more. This, however, was so slight that it was difficult to observe on every occasion. It is,nevertheless, bourne out by the decrease in indicated efficiency as discussed earlier.

In the supercharged operation the exhaust was so clean in every instance that any visual observation of changes in color was impossible. The trends of the indicated efficiency in the supercharged operation (almost constant) also tend to show that combustion in this mode was scarcely affected by water injection.

CHAPTER IV

DISCUSSION OF RESULTS

This chapter attempts to analyze and explain the results presented in the previous chapter as well as offer some thoughts on future applications and recommendations for further investigations.

A. MODELS OF INTERNAL COOLING

Examination of the results has led the author to consider two models for the phenomenon of internal cooling. Both models will be referred to at various times during the discussion to point out their applicability or non-applicability as the case may be. In as much as the entire process of internal cooling in a diesel engine is a complicated situation, and since this investigation was limited in several areas, the author does not conclude that one model represents the process better than the other. Further study of internal cooling is required to do this.

(1) Wet Cylinder Wall Approach

In this view it is assumed that the water injected into the manifold and fed into the combustion chamber acts as a heat absorbing medium only on the combustion surfaces. Normal combustion occurs, except for a very small amount of dilution effect due to manifold evaportation. The internal cooling occurs as the water, deposited wet on the combustion chamber surfaces, evaporates due to the normal heat of

combustion released during the combustion period. The evaporated water is then discharged with the exhaust, carrying with it some of the heat normally transferred through the walls of the combustion surfaces to the external cooling circuit.

(2) Homogeneous Mixture Approach.

In this view it is assumed that the injected water is perfectly mixed with the air in the intake charge, and no wetting of the combustion chamber surfaces occurs during the cycle. The actual internal cooling commences in the manifold and during the intake stroke as the water vaporizes from its atomized form and cools the inlet charge. The majority of the internal cooling takes place during the combustion process. The water absorbs heat constantly during each event, preventing normal peak combustion temperatures from being reached and thus allowing less heat to be transferred through the combustion surfaces to the external cooling circuits. The energy absorbed by the water is discharged with the exhaust.

B. HEAT TRANSFER

(1) Correlations Established.

Rather than attempt to analyze each of the individual curves as presented in the previous chapter, the author felt that to provide an overall view of the phenomenon of internal cooling by manifold water injection it would be more meaningful to consider all of the heat transfer data in some sort

of collected form. Figure IV-1 presents all the data for the supercharged operation in one series of curves. In this figure, the decrease in total heat rejected by the cooling system is plotted against the mass flow rate of injected water for each of four approximate fuel air ratios. Examination of the curves shows that the reduction in heat rejected appears to be dependent upon not only the water flow rate as expected, but also the F/A ratio. This can be seen from the individual heat rejection curves presented in Chapter III, but it is considerably clearer from the collected data plot.

If the data presented in Figure IV-1 is non-dimensionalized and the parameter changed, another interesting feature is observed. The effectiveness of the internal cooling appears to reach a maximum at an F/F_{cc} of about 0.60. This is illustrated in Figures IV-2A and IV-2B which depict the trend of the ratio of decreased heat rejected and fuel energy input ($\Delta\dot{Q}/E$) versus F/F_{cc} for all values of W/F. Due to the small amount of data gathered and the considerable scatter when reduced in this fashion, it is difficult to determine whether the maximums shown shift with F/F_{cc} for the various W/F ratios, and hence the curves only show the rough trends of the data.

Due to this problem, an unquestionable explanation of the maximum is impossible. One reason may be that at the

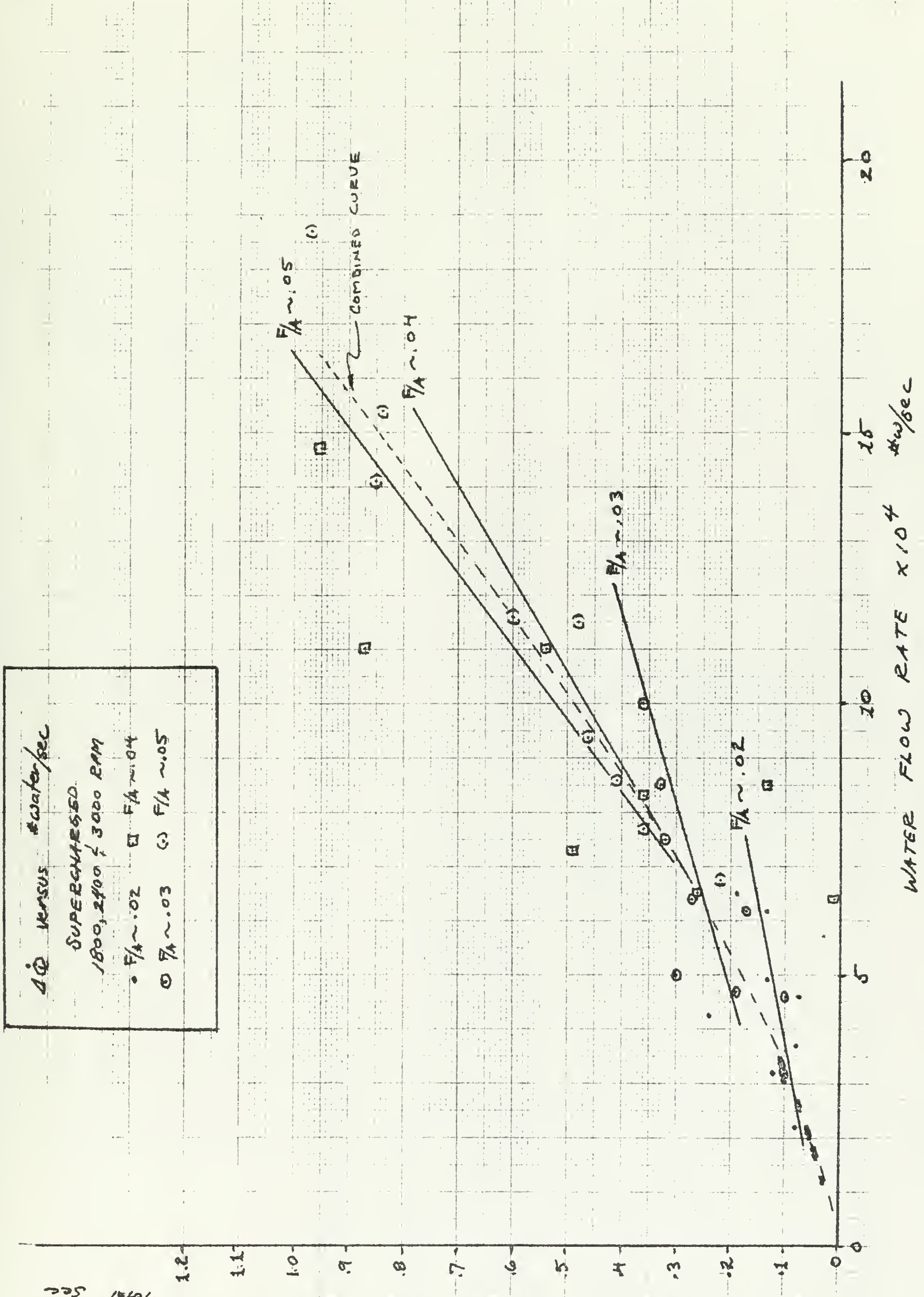
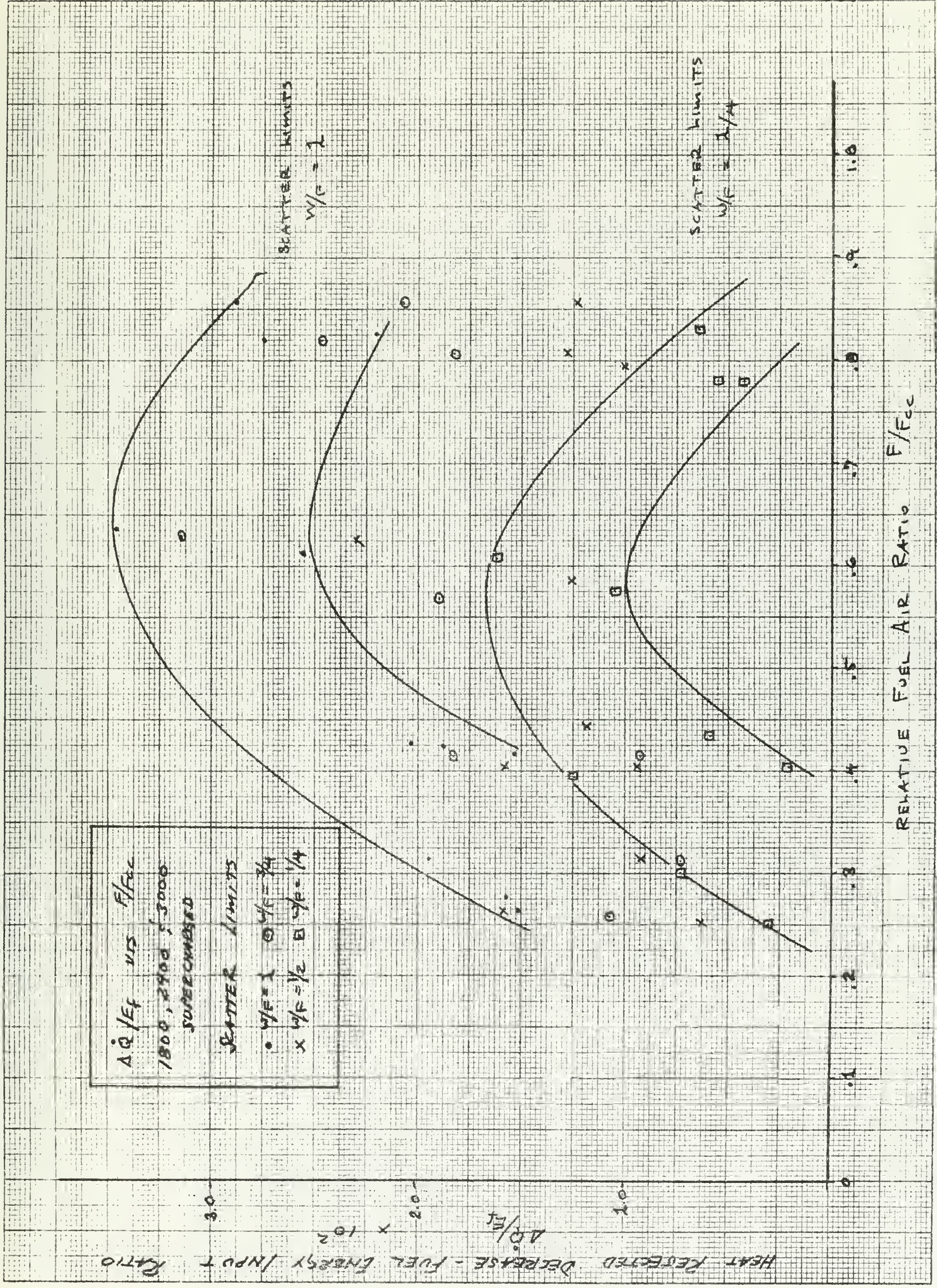
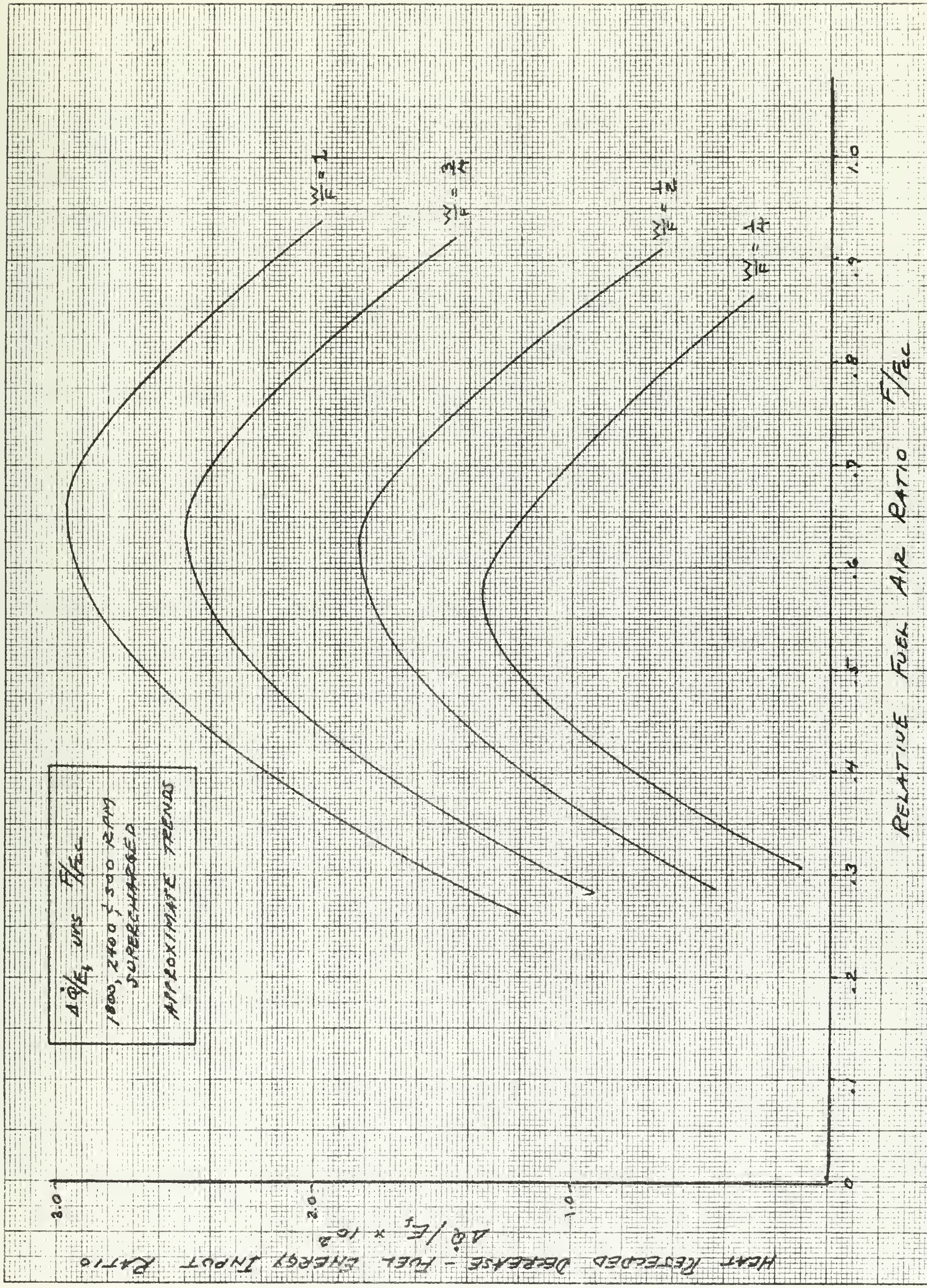

ΔQ̇ versus #water/sec
SUPERCHARGED
1800, 2400 & 3000 RPM
F/A ~ .02
F/A ~ .04
F/A ~ .03
F/A ~ .05
COMBINED CURVE
F/A ~ .05
F/A ~ .04
F/A ~ .03
F/A ~ .02
WATER FLOW RATE x 10⁴ #w/sec
DECREASE IN TOTAL HEAT REJECTION ΔQ̇total BTU/sec

FIGURE IV - 2A


ΔQ/Ef vs F/Fcc
1800, 2400 & 3000
SUPERCHARGED
SCATTER LIMITS
• W/F = 1 ⊙ W/F = 3/4
× W/F = 1/2 ⊡ W/F = 1/4
SCATTER LIMITS
W/F = 1
SCATTER LIMITS
W/F = 1/4
RELATIVE FUEL AIR RATIO F/Fcc
HEAT REJECTED DECREASE - FUEL ENERGY INPUT RATIO
ΔQ/Ef × 10^2
0
.1
.2
.3
.4
.5
.6
.7
.8
.9
1.0
2.0
3.0


ΔQ̇/Ė_f vs F/F_EC
1800, 2400 & 3500 RPM
SUPERCHARGED
APPROXIMATE TRENDS
W/F = 1
W/F = 3/4
W/F = 1/2
W/F = 1/4
RELATIVE FUEL AIR RATIO F/F_EC
0
.1
.2
.3
.4
.5
.6
.7
.8
.9
1.0
HEAT REJECTED DECREASE - FUEL ENERGY INPUT RATIO
ΔQ̇/Ė_f × 10^2
1.0
2.0
3.0

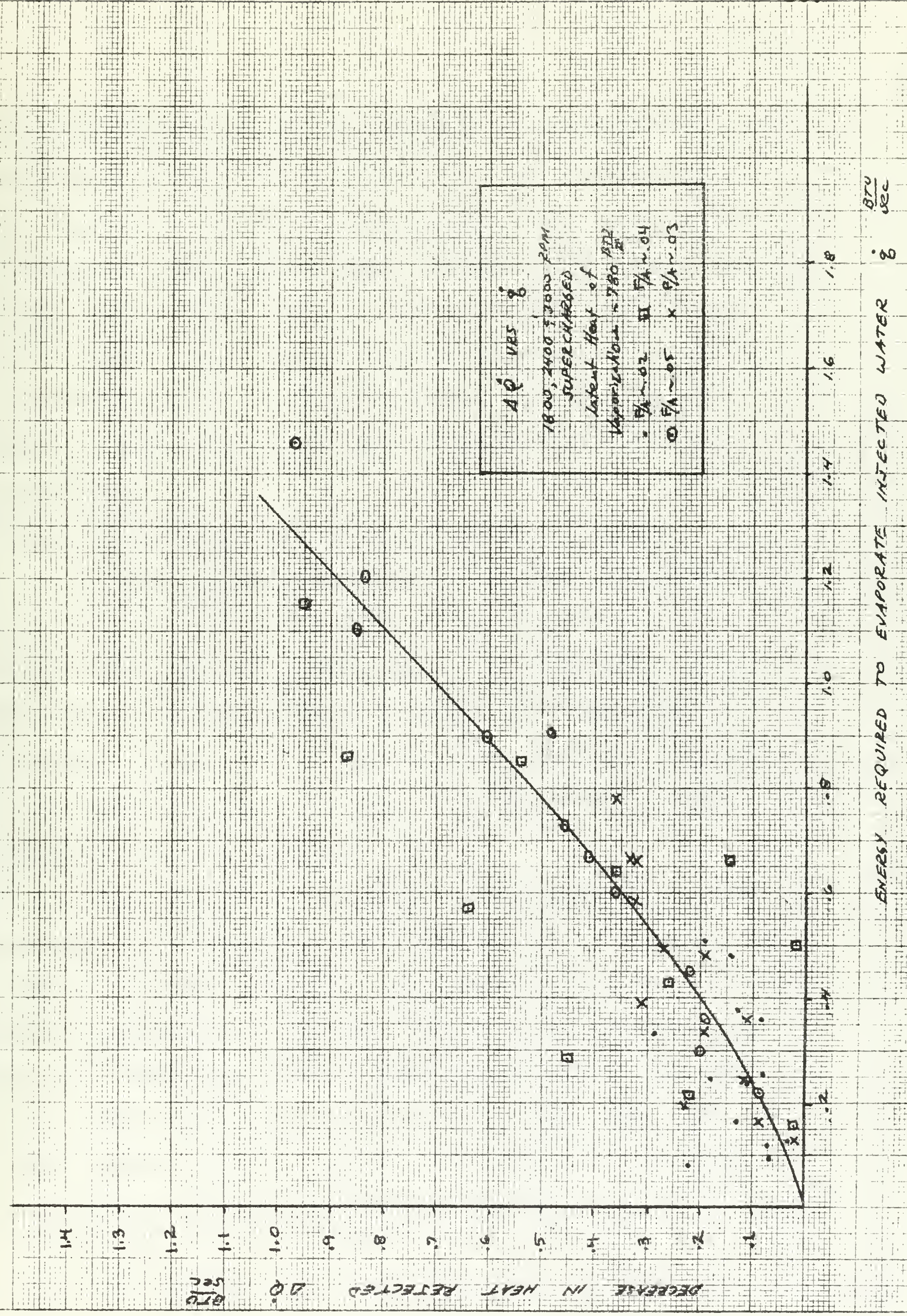
lower F/A ratios all the water is never completely vaporized and as the F/F_{cc} increases more and more water becomes vaporized until F/F_{cc} of 0.60 is reached. At this point all the water is vaporized and the latent heat of vaporization no longer controls the heat absorption by the water. The specific heat of steam controls this absorption instead. Since the specific heat stage would absorb considerably less heat, and since more heat is being liberated during the combustion at the higher F/F_{cc}, the effectiveness of the internal cooling decreases.

If this were the case, the curves should reach their maximum values at higher and higher values of F/F_{cc} for increasing values of W/F ratios, since as W/F increases more and more water is available. This feature of a shift in maximums was not obvious, but perhaps more closely gathered data would illustrate it.

The above reasoning for the maximum effectiveness of internal cooling at F/F_{cc} at about 0.60, if correct, could apply to either model proposed earlier. It is difficult to imagine, however, that with the small amounts of water injected, total evaporation would not always occur, even at low values of F/F_{cc}.

Another interesting relationship is shown in Figure IV-3. Here the decrease in heat rejected is plotted against the energy necessary to evaporate the injected water at all data points. In this curve no effect of F/A ratio was noticed.

FIGURE IV - 3


ENERGY REQUIRED TO EVAPORATE INJECTED WATER $\dot{q}$ $\frac{BTU}{sec}$
DECREASE IN HEAT REJECTED $\Delta\dot{Q}$ $\frac{BTU}{sec}$
$\Delta\dot{Q}$ VRS $\dot{q}$
1800, 2400 & 3000 RPM
SUPERCHARGED
Latent Heat of
Vaporization = 780 BTU/lb
· F/A ~ .02 ⊡ F/A ~ .04
⊙ F/A ~ .05 × F/A ~ .03

The correlation between the decrease in heat rejected by the cooling system and the latent heat of the injected water is remarkable. As can be seen there is only slightly less internal cooling than available solely by the latent heat of vaporization. This could be due to partial vaporization of the water in the intake manifold as well as the over-simplified assumption that the latent heat was constant and at a value corresponding to a mean cylinder pressure of 400 psia.

(2) Prediction of the Degree of Internal Cooling

From the proceeding curve of the relationship between the amount of internal cooling and the latent heat of vaporization of the injected water, it is possible to be at least relatively correct to assume that the degree of internal cooling will be equivalent to the heat required to evaporate the injected water. This provides a simple solution on the gross scale.

Another method is possible if the Homogeneous Mixture Approach is considered. Here, recall that in altering the process of normal combustion, the water prohibits the normal peak temperatures from being reached in the cylinder. As such the mean effective gas temperature, T_g, is decreased. Using empirical formulas developed by Taylor and Taylor in their analysis of heat transfer in engines, it is possible to determine an expression between the change in mean effective

gas temperature, ΔT_g, and the W/F ratio. Consider the following expression:

$$\frac{\dot{Q}}{A(T_g - T_c)} = \frac{10.4\ K_g}{b}\ (R_{e\ gas})^{.75}$$

where

$\dot{Q}$ = BTU/time transferred through the walls of the combustion chamber of area A

$R_{e\ gas}$ = the gas side Reynolds number $\frac{Gb}{\mu g_o}$ with G being the mass flow rate of the gases divided by the piston area

$(T_g - T_c)$ = temperature difference between the mean gas temperature and the mean coolant temperature

b = characteristic length, conveniently taken as the engine's bore.

K_g = the thermal conductivity of the gases in the cylinder; conveniently taken the conductivity of the inlet air.

Assuming that the values of K_g and μ are not significantly altered by manifold water injection, it can be seen that $\dot{Q}$ decreases with a decrease in mean effective gas temperature. Since $\dot{Q}$ does in fact decrease with increasing values of W/F, the mean gas temperature decreases with water injection.

Other than the water flow rate, the only operating variables that effect the mean gas temperature in an important way are the fuel air ratio, the inlet temperature and exhaust pressure. Thus, since the inlet temperature after water injection and the exhaust pressure ratios should affect the mean gas temperature, there by substantiating the trends

FIGURE IV - 4A

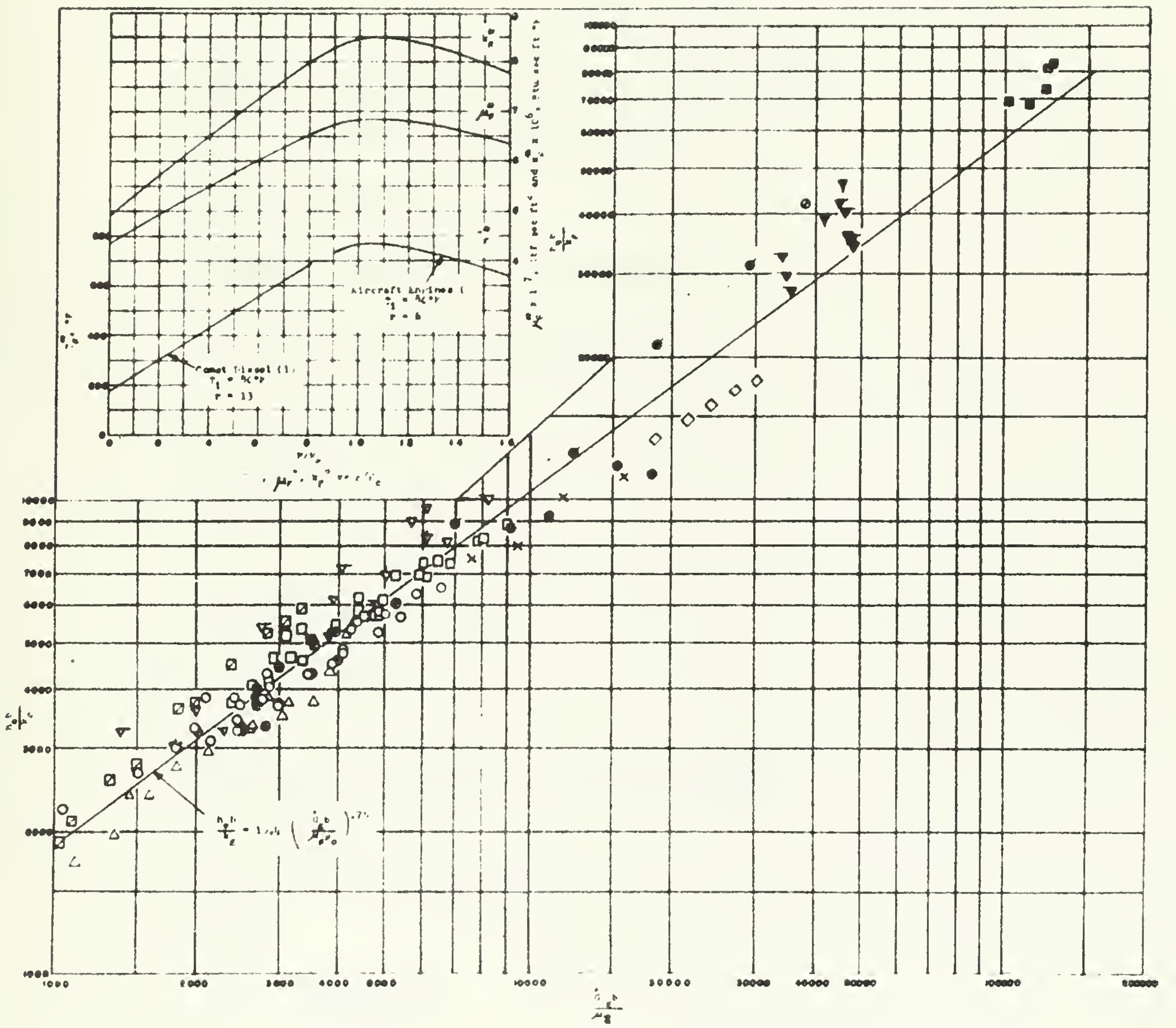


FIGURE IV - 4B

$h_e = \dot{Q}/A_p(T_g - T_c)$

$\dot{Q}$ = heat to jackets plus heat to oil minus heat of friction when such data were available. Otherwise $\dot{Q}$ = heat to jackets only

A_p = piston area

T_g = mean effective gas temperature

T_c = coolant outlet temperature

T_i = temperature of gases in inlet manifold

$T_g = T_g{}^* + \Delta T_g$

$T_g^* = T_g$ when T_i = 80%

$\Delta T_g = .35(T_i - 80)$

$\dot{G}_g$ = mass flow of fuel and air/time x piston area

r = compression ratio

b = bore

$g_o = 32.2 \dfrac{\text{lbm ft.}}{\text{lbf}\ \text{sec}^2}$

μ_g = viscosity of air at T_g

k_g = thermal conductivity of air at T_g

F = fuel-air ratio

F_c = chemically correct fuel-air ratio

depicted in Figure IV-1.

From the heat transfer data and Taylor and Taylor's correlations, shown for illustration purposes in Figure IV-4, curves of T_g versus relative fuel air ratio (F/F_{cc}) were drawn for all W/F ratios. From these curves shown in Figure IV-5, curves of ΔT_g versus F/F_{cc} were constructed. ΔT_g represents the change in T_g from the no water injection condition to the particular W/F condition under consideration. Figure IV-6 shows ΔT_g versus F/F_{cc}. From Figure IV-6 the following expression, which shows the relationship between the internal cooling effect and the W/F and F/A ratios, was determined:

$$\Delta T_g = \frac{W}{F} \left(135 \frac{F}{F_{cc}} - 16\right)$$

Although there was considerable scatter of the reduced heat transfer data and although a rather limited amount of data was obtained, the above relationship is felt to be at least indicative of the trend of the decrease in the mean gas temperature. The reduction of the data to obtain the correlation is relatively insensitive to experimental error, that is it neither attenuated nor reduced the error.

It is recommended that all further results be incorporated into and compared with this expression in an effort to improve the accuracy of the relationship.

FIGURE IV - 5

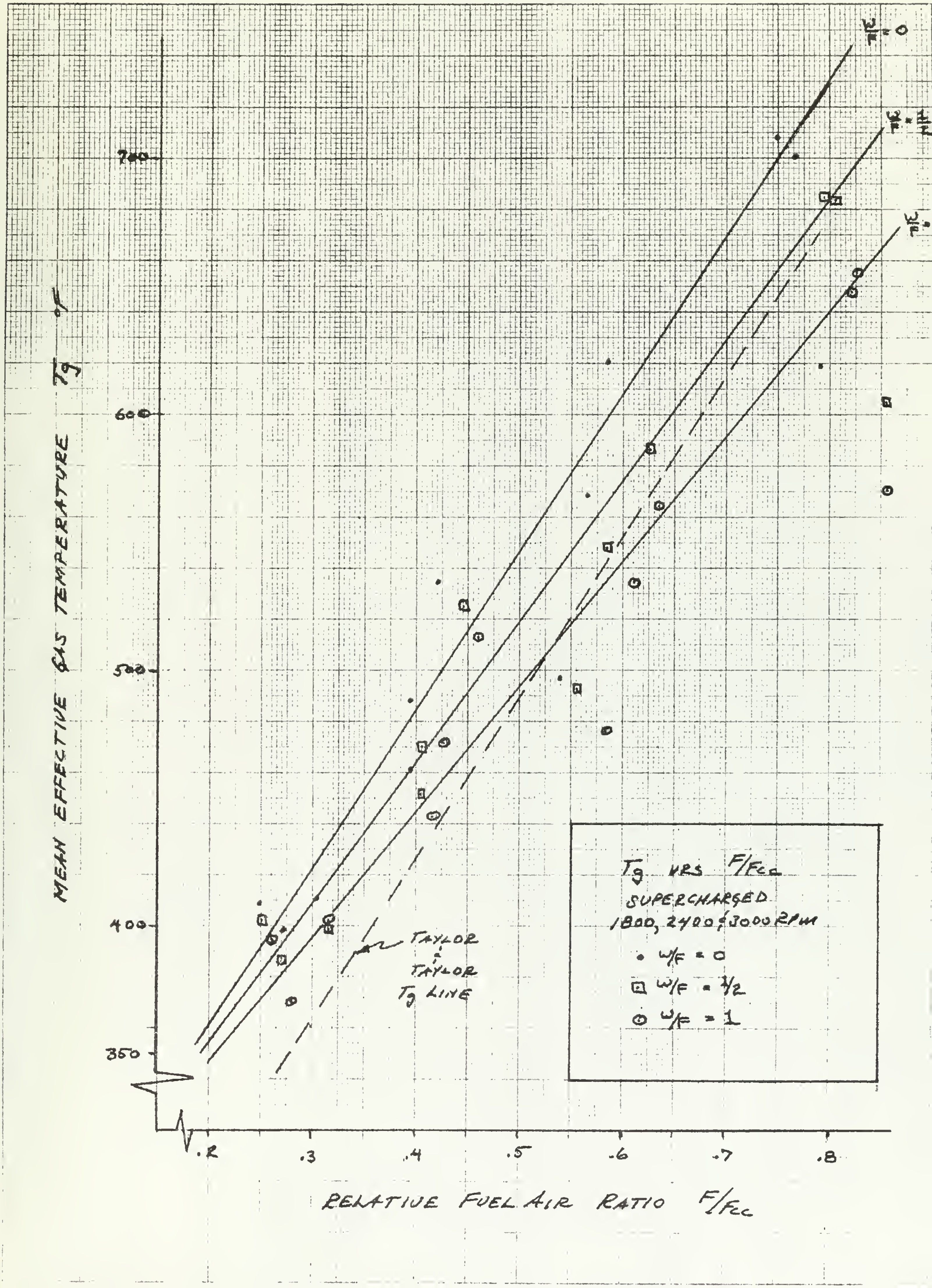
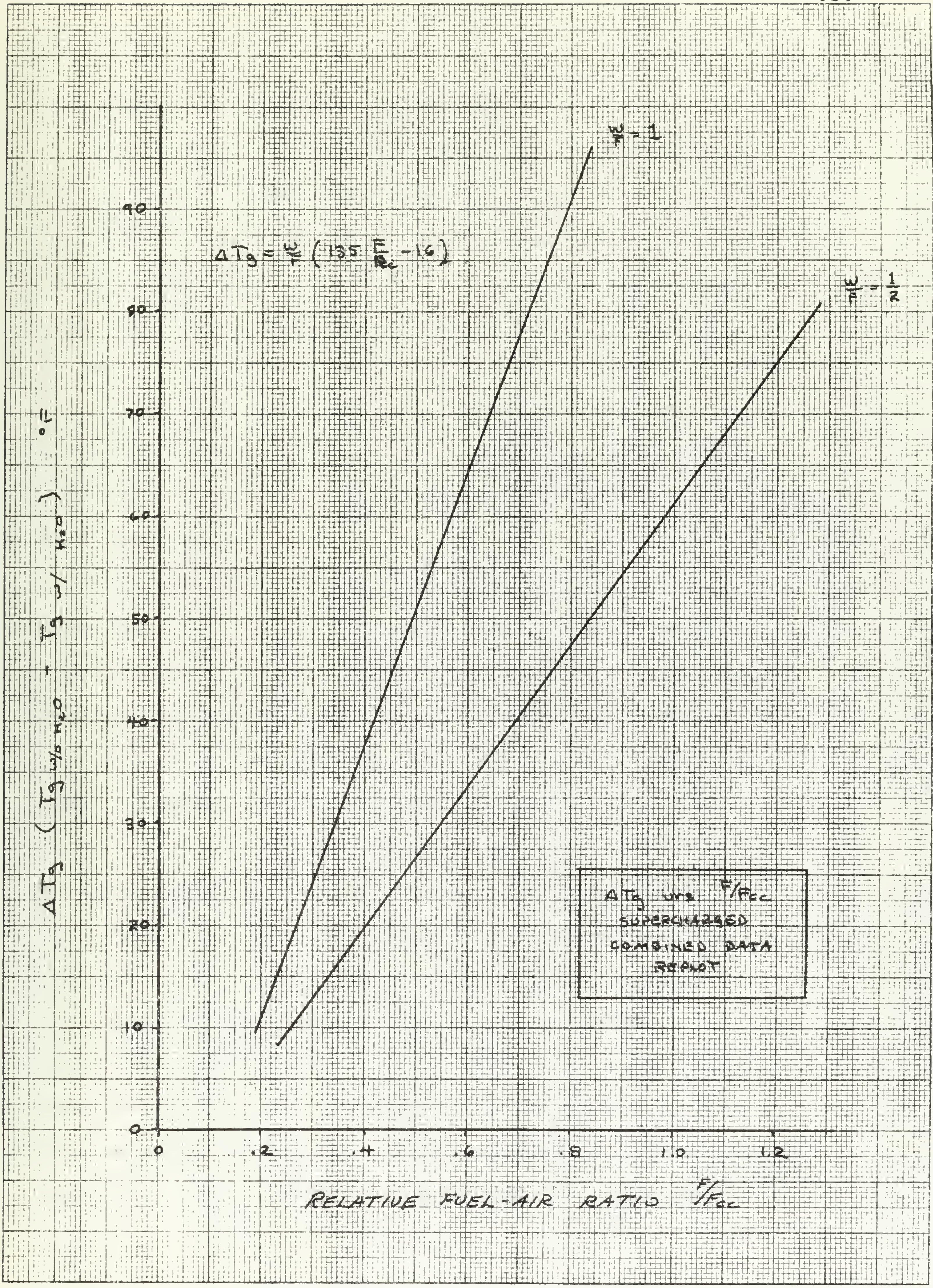

MEAN EFFECTIVE GAS TEMPERATURE Tg °F
700
600
500
400
350
.2
.3
.4
.5
.6
.7
.8
RELATIVE FUEL AIR RATIO F/Fcc
W/F = 0
W/F = 1/2
W/F = 1
TAYLOR & TAYLOR Tg LINE
Tg VRS F/Fcc
SUPERCHARGED
1800, 2400 & 3000 RPM
• W/F = 0
⊡ W/F = 1/2
⊙ W/F = 1

FIGURE IV - 6



$\frac{W}{F} = 1$
$\Delta T_g = \frac{W}{F}\left(135 \frac{F}{F_{cc}} - 16\right)$
$\frac{W}{F} = \frac{1}{2}$
ΔT_g ($T_{g\,w/o\,H_2O}$ - $T_{g\,w/\,H_2O}$) °F
0
10
20
30
40
50
60
70
80
90
.2
.4
.6
.8
1.0
1.2
RELATIVE FUEL-AIR RATIO F/Fcc
ΔT_g vrs F/Fcc
SUPERCHARGED
COMBINED DATA
REPLOT

C. POWER AND VOLUMETRIC EFFICIENCY

Due to the test procedure selected, no evaporative cooling of the intake charge by the manifold water injection was allowed. As such the volumetric efficiency decreased and the power fell off slightly. In actual field use of water injection, evaporative cooling would occur as no operator would ever think of heating the intake air charge except for ease of starting in cold climates. Thus the volumetric efficiency would actually increase with water injection, and as has been shown in reference (9), the power would increase.

The advantage of evaporative cooling effects appear especially well suited to highly supercharged diesels utilizing exhaust turbochargers with high pressure ratios. As the compressed air discharged from the turbocharger is considerably higher in temperature than ambient air, substantial charge cooling by water injection would occur. Thus, by utilizing water injection not only would the power be increased by increasing the volumetric efficiency, but also the internal cooling effects would result in less heat being rejected to the cooling system. Furthermore, it is expected that with manifold water injection the exhaust energy would increase as the W/F ratio due to the internal coolant absorbing heat in the combustion chamber and being expelled with the exhaust. This increase in exhaust energy would seem to benefit the turbocharger. It is questionable,

however, whether this increase in exhaust energy would be available. Undoubtedly some of this energy would be in the chemcial form and not capable of being utilized by the turbocharger.

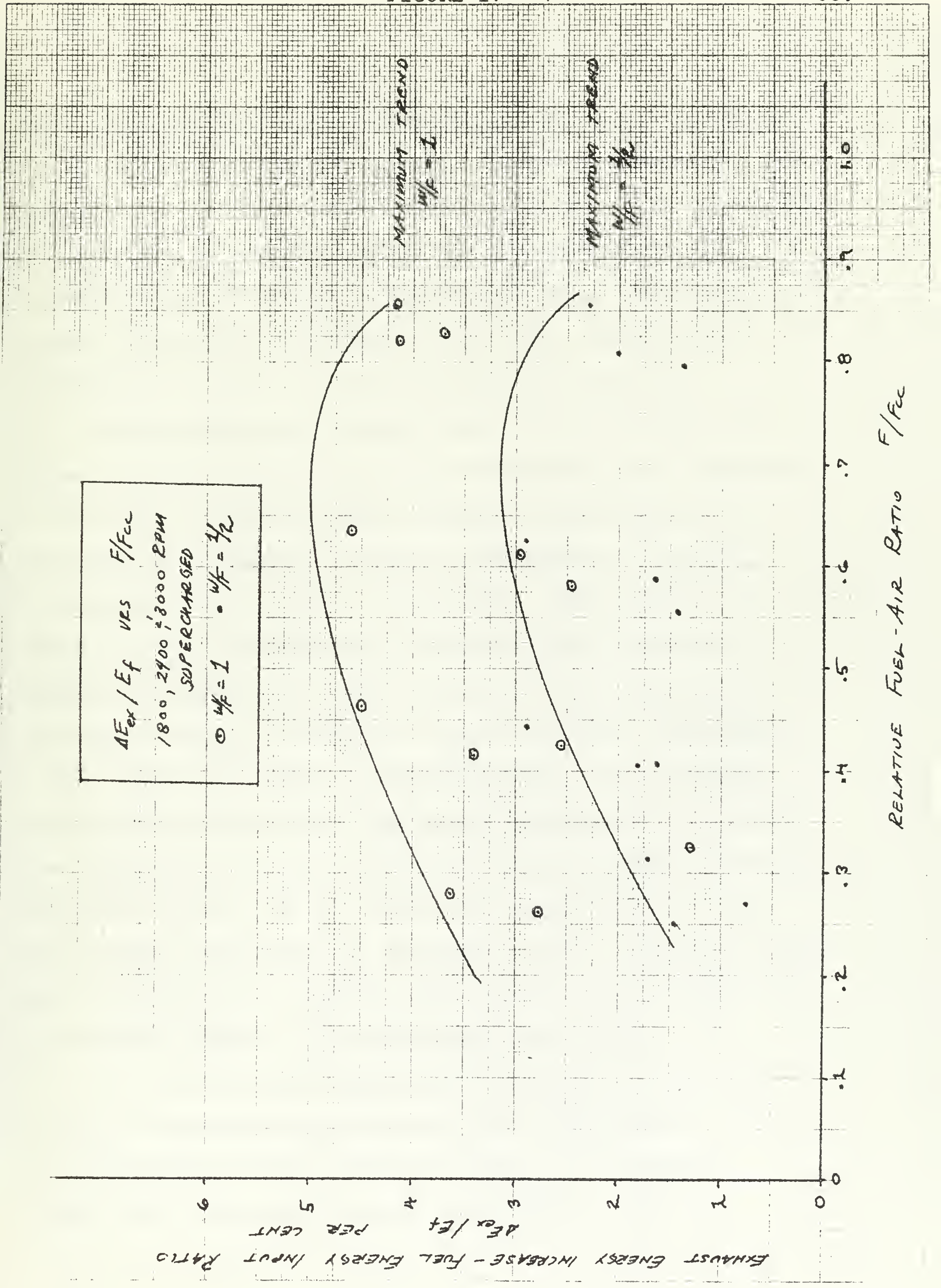
In illustration of the increase in exhaust energy of all forms with increasing W/F ratio, Figure IV-7 shows the increase in exhaust energy non-dimensionalized by the energy input by the fuel, $(\Delta E_{ex}/E_f)$, plotted against F/F_{cc} for W/F ratios of 0.50 and 1.00. The increase in exhaust energy was calculated assuming an actual indicated cycle approach where the energy of the fuel appears as exhaust, power and cooling. Since all values except the exhaust were known, the exhaust energy was readily available. As can be seen from the graph, there was considerable scatter of the data and hence only maximum trend lines are faired in. Note that a maximum appears corresponding to F/F_{cc} of about .06. This is a result of the maximum effectiveness of internal cooling at F/F_{cc} of about .06 discussed earlier.

In further investigations it is recommended that charge cooling be allowed and that exhaust temperature be recorded.

D. INDICATED EFFICIENCY AND COMBUSTION

Depending upon the particular model selected, the effect of water injection on the efficiency and combustion is markedly different.

As regards the Wet Cylinder Wall Approach, the effect on both the efficiency and the combustion should be nil. Since


ΔE_ex / E_f VRS F/F_cc
1800, 2400, 3000 RPM
SUPERCHARGED
⊙ W/F = 1 • W/F = 1/2
MAXIMUM TREND W/F = 1
MAXIMUM TREND W/F = 1/2
RELATIVE FUEL-AIR RATIO F/F_cc
EXHAUST ENERGY INCREASE - FUEL ENERGY INPUT RATIO
ΔE_ex / E_f PER CENT

this study showed slight decreases in efficiency, it appears as if this model is not accurate. However, since all the water injected would not necessarily be on the combustion surfaces due to evaporation off the manifold pipe wall, a small amount of charge dilution would occur causing a slight hinderance to combustion and thus a decrease in efficiency as noted in the unsupercharged operation.

If the Homogeneous Mixture Model is considered, the primary effect of the water injection appears to be dilution of the active charge in the cylinder. This results in a decrease in the maximum cylinder temperatures as shown by a decrease in the mean gas temperature. Additionally, as shown by the unsupercharged operation, this decrease in maximum temperatures probably resulted in the increase of the delay period to combustion requiring slight advancement of the injection timing for optimum power. It is probably more accurate to say that the water injection did not allow the temperature to rise as fast as normal thereby prolonging the delay period. In any case, the end result is the same; the maximum temperature is decreased and the injection timing must be advanced for optimum power. Since normal combustion is somewhat impaired, the efficiency falls off.

In as much as the supercharged operation did not show a similar decrease in efficiency, nor a requirement for advancement of the injection timing, the dilution effect does not appear to be as pronounced. This

is logical in that the peak pressures are higher tending to offset the hinderance to combustion caused by diluation with water.

As mentioned in Chapter III the influence of the water injection on the combustion process is difficult to analyze in this investigation. It is therefore recommended that any further study of this area include indicator apparatus to assist in the analysis.

E. FUTURE APPLICATIONS

Future applications of manifold water injection in a diesel engine hinge on one item - economic advantage. Although the economics as such cannot be discussed since this area was not considered in this study, several distinct applications of water injection were made apparent during the course of the project. The major conclusion to keep in mind in this regard is that water injection decreases the heat rejected to the cooling system without significant adverse affect on the power output or efficiency of the engine.

First of all, if engine heat rejection to the cooling system represents a significant problem in special high power applications, internal cooling may provide the answer. Results of this study show that in a percentage comparison of heat rejection versus power output the decrease in heat in rejected to the cooling system is typically twice as much as the decrease in power output at the higher F/A ratios

where an engine of this type would be expected to operate. With the help of evaporative cooling of the inlet air by the water spray in the intake manifold this ratio would be expected to increase. This, coupled with the decrease in thermal stresses due to possible reduction in cylinder gas temperatures, could provide incentive for actual field usage of water injection.

The most likely chance of success for an internal coolant, however, is in the possible reduction in the emissions of oxides of nitrogen. Previous investigations have shown the dominant influence of combustion temperature on the theoretical nitric oxide concentrations in spark ignition engines and it is reasonable to conclude that the same effect holds true for compression ignition engines. If the mean gas temperature decreases with water injection, it is expected that this would tend to decrease the formation of nitric oxide during combustion. To determine the extent to which emissions of these oxides of nitrogen are reduced and to check the validity of the Homogenous Mixture Model, it is recommended that further investigations including exhaust gas analysis be conducted.

An interesting variation of water injection for internal cooling should not be overlooked in future studies. This particular study utilized manifold water injection and was thus limited to "injection" of water during the normal in-take stroke. Perhaps more efficient internal cooling, less of

a decrease in power, or more of a reduction in nitric oxide formation would result if the water were directly injected into the cylinder by means of an injector with controllable injection timing and spray patterns. If this method were employed, the optimum injection timing and spray pattern for the desired result could be determined.

CHAPTER V

CONCLUSIONS AND RECOMMENDATIONS

In summary of this study on internal cooling of high speed diesel engines with manifold water injection, the conclusions and recommendations appearing in the text of this report are listed.

A. CONCLUSIONS

1. Manifold water injection decreases the heat rejected to the cylinder head. Based on this result two models of internal cooling are proposed.
2. To assist in the prediction of the amount of internal cooling, two estimating methods are presented. The first is based upon the latent heat of vaporization of the injected water and the second is based upon the degree of reduction in the mean effective gas temperature with water injection.
3. Volumetric efficiency and power decreased mainly due to the maintaining of a constant inlet air temperature. Although not permitted, there was a strong tendency for the inlet air to be cooled by the manifold water injection system.
4. Indicated efficiency was somewhat impaired, primarily in the unsupercharged operation. Slight advancement

of injection timing was required in this mode indicating a prolonging of the delay period of combustion.

5. High speed diesels do not balk at water injection up to rates equal to the fuel flow rate.

6. In the quest for cleaner exhaust, manifold water injection appears to fail. There are indications, however, that reductions in nitric oxide emissions are possible with water injection.

B. RECOMMENDATIONS

1. Further studies should be conducted on manifold water injection in diesel engines to determine:

   (a) The correctness of the models presented

   (b) The accuracy of the presented prediction estimates for internal cooling

   (c) The degree of increased power due to charge cooling effects with emphasis on the limits involved

   (d) More complete combustion data

   (e) Maximum W/F limit

   (f) Degree of reduction in emission of nitric oxide

2. Studies on direct injection of water into the cylinder with variable injection timing and controllable spray pattern should be undertaken.

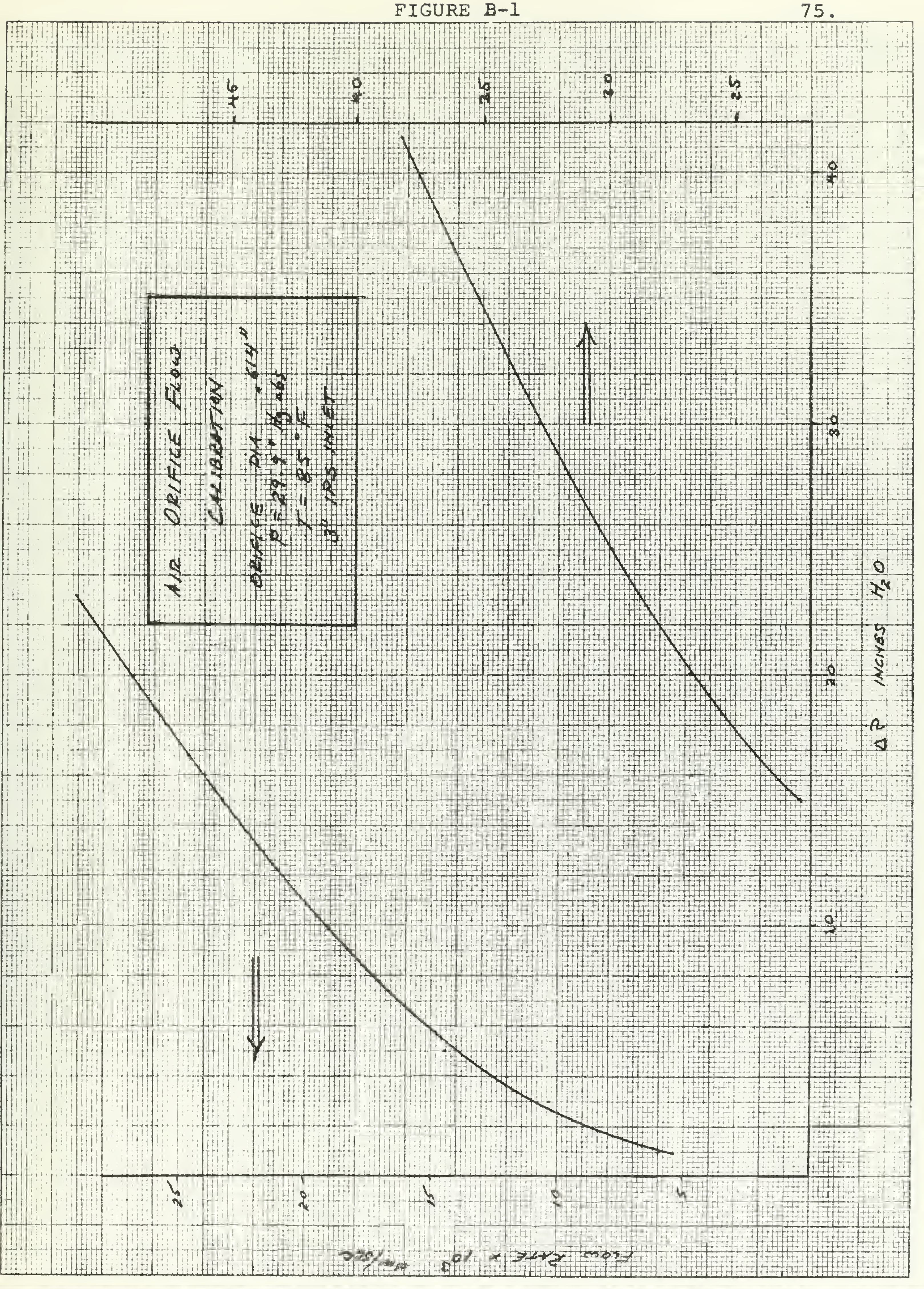
APPENDIX A

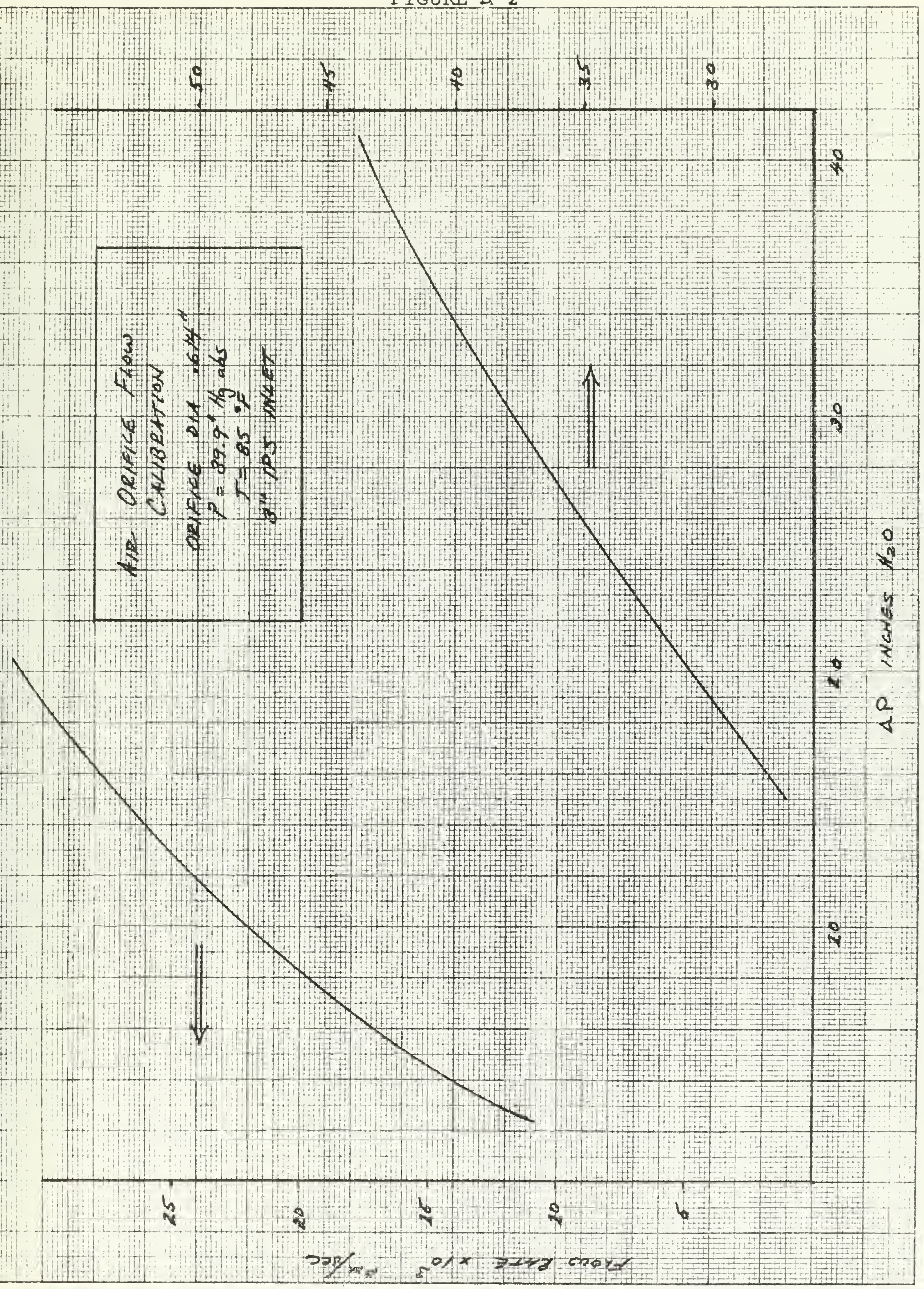
NOMENCLATURE

ΔE_{ex}	change in the exhaust energy from the no water condition to the particular W/F under consideration.
E_f	fuel energy input BTU/sec
F/A	fuel-air ratio
F/F_{cc}	fuel-air ratio relative to stoichiometric fuel-air ratio
IMEP	Indicated Mean Effective Pressure psi
$_{i}$	Indicated Efficiency
$_{vol}$	Volumetric Efficiency
$\dot{Q}$	heat flow rate BTU/sec
$\Delta\dot{Q}$	change in heat flow rate from the no water condition to the particular W/F under consideration BTU/sec
$\dot{q}$	heat flow rate necessary to vaporize the injected water BTU/sec
T_g	mean effective gas temperature°F
ΔT_g	change in mean effective gas temperature from the no water condition to the particular W/F under consideration
W/F	water-fuel ratio

APPENDIX B

CALIBRATION CURVES


AIR ORIFICE FLOW
CALIBRATION
ORIFICE DIA .614"
P = 29.9" Hg abs
T = 85 °F
3" IPS INLET
ΔP INCHES H2O
FLOW RATE × 10^3 #/SEC
10
20
30
40
5
10
15
20
25
45
40
35
30
25


AIR ORIFICE FLOW
CALIBRATION
ORIFICE DIA .614"
P = 39.9" Hg abs
T = 85 °F
8" IPS INLET
ΔP INCHES H2O

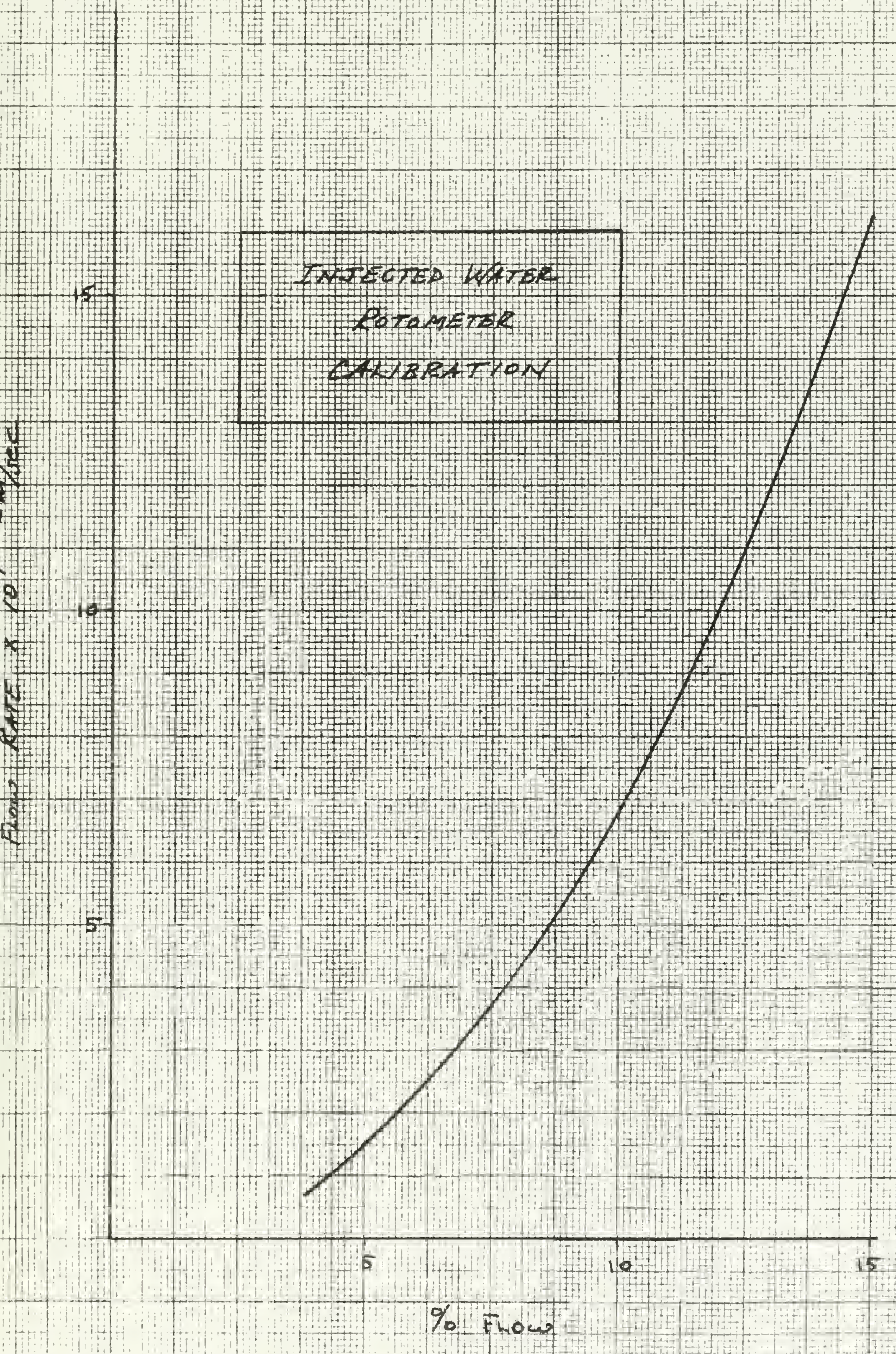

INJECTED WATER
ROTOMETER
CALIBRATION
FLOW RATE x 10⁴ #m/SEC
15
10
5
5
10
15
% FLOW

FIGURE B - 4

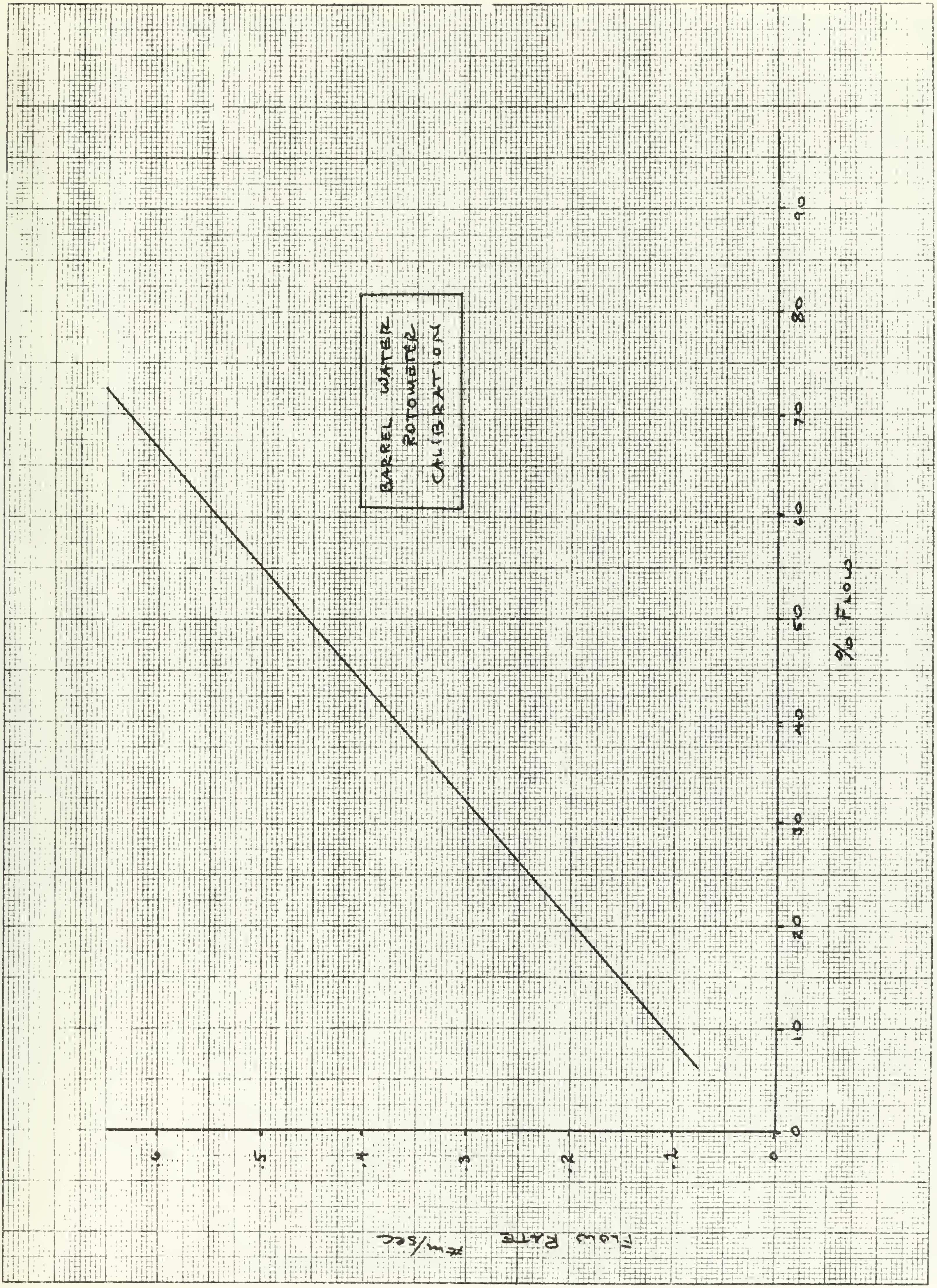
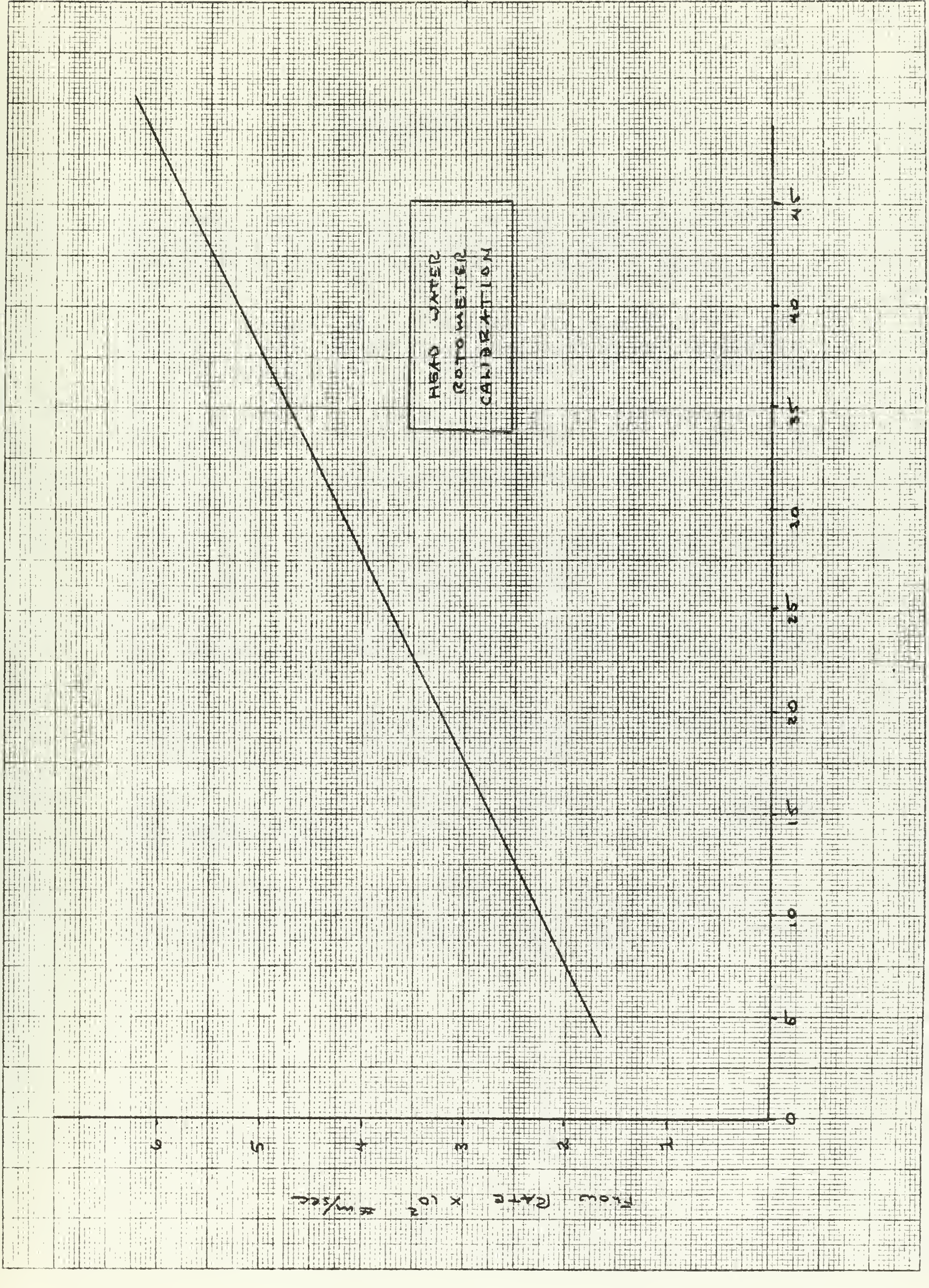

BARREL WATER
ROTOMETER
CALIBRATION
% FLOW
0
10
20
30
40
50
60
70
80
90
FLOW RATE
#m/SEC
0
.1
.2
.3
.4
.5
.6

FIGURE B - 5


HEAD WATER
ROTOMETER
CALIBRATION
0
1
2
3
4
5
6
5
10
15
20
25
30
35
40
45
Flow Rate x 10² #m/sec

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POSTSCRIPT

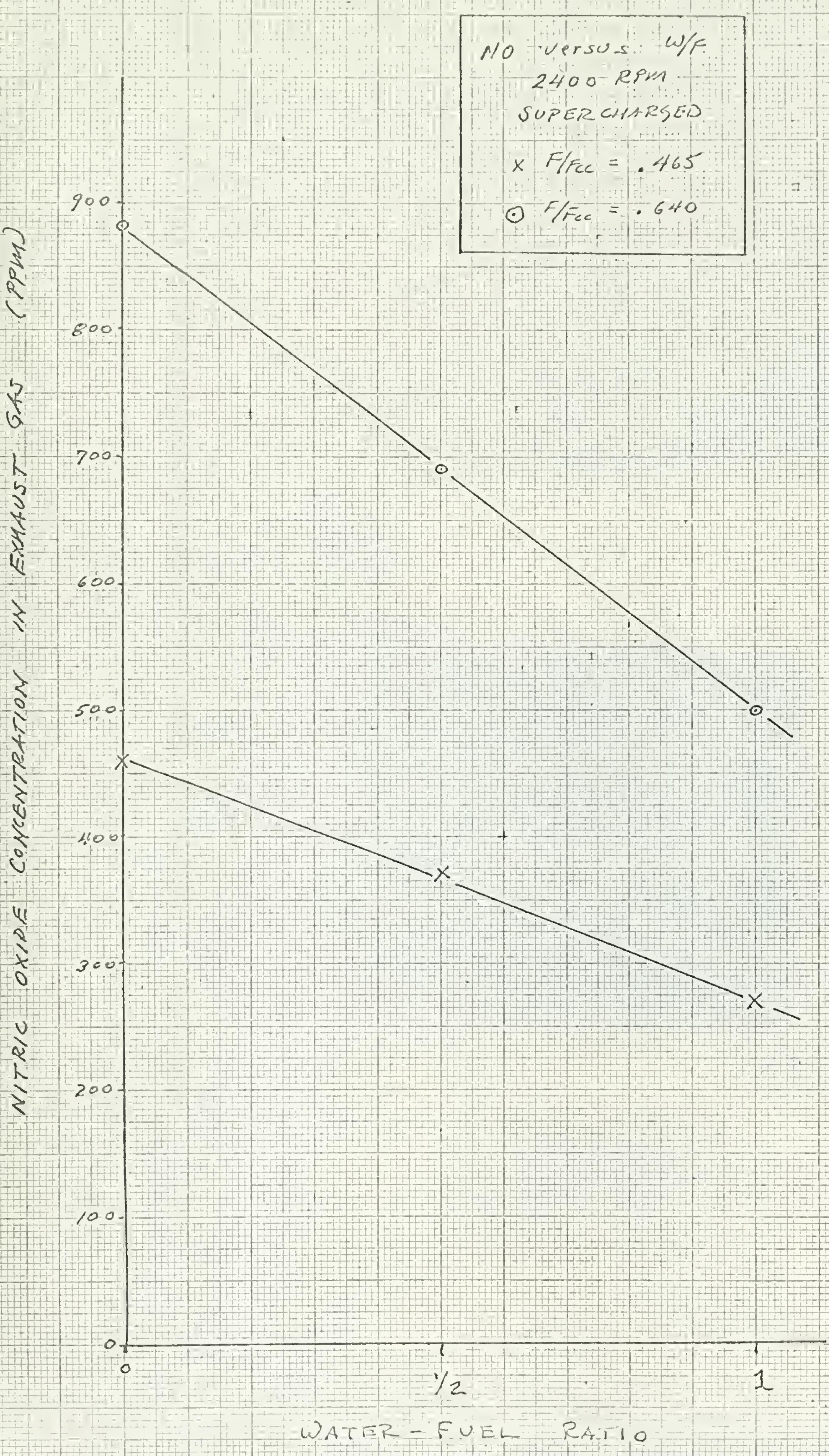
This study ended leaving one major question unanswered - that of a possible reduction in emissions of nitric oxide (NO) with manifold water injection. To determine if in fact a reduction in these emissions occurred, a short test run utilizing an infra-red exhaust gas analyzer was conducted.

The speed of 2400 RPM was selected for this run. Two fuel-air ratios, F/F_{cc} of 0.465 and 0.640, and three water fuel ratios, W/F of 0, 1/2 and 1, were investigated.

The results of the short study showed that there was a reduction in NO emissions of approximately 40%. This is shown in Figure P - 1.

In view of the foregoing it is strongly recommended that further studies be conducted to determine more information on this vital aspect of engine operation with manifold water injection.

FIGURE P - 1


NO versus W/F
2400 RPM
SUPERCHARGED
× F/Fcc = .465
⊙ F/Fcc = .640
900
800
700
600
500
400
300
200
100
0
NITRIC OXIDE CONCENTRATION IN EXHAUST GAS (PPM)
0
1/2
1
WATER - FUEL RATIO

Thesis 127350
H1627 Hamilton
Internal cooling of a high speed supercharged diesel engine by manifold water injection.